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PRODUCT ENGINEERING

VOLUME 5 NUMBER 6

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Coming Issues

In light-weight construction there are three materials that are playing leading rôles. They are alloy steels, aluminum alloys and magnesium alloys. Each of these will be the subject of a major article.

Economy is usually the reason given for using cold-drawn steel sections. Cost is always a factor but often there are other more important reasons. Find out about them in the July number.

How to design cams for maximum speed will be the subject of another article in the July number.

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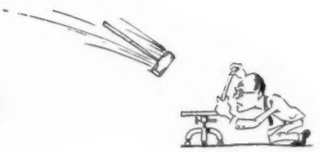
Intimate Correspondence



Fine—But

PRODUCT ENGINEERING
NEW YORK

We wish to express our appreciation for the publicity given to the National Alliance of the Art & Industry's Designers Exhibition at the RCA Building. The designers have put up a very presentable show for the first year, and the publicity with which you have supplied them, should be a real means of attracting the right people in industry to the show.

We note, however, that while four of the plastic manufacturers are exhibiting at the show, your article on the show gives credit generically to all things as "Bakelite." Bakelite, as you must know, is the trade name of but one manufacturer.

For instance, the Coca-Cola stand is made of Formica, and while Formica laminated stock is made partially of Bakelite resin, it is also made of paper, cloth and other raw material.

H. S. SPENCER, *Advertising Manager*
General Plastics, Inc.,

WE stand corrected, and of course we know that it is careless to apply the term "Bakelite" to all plastics as a generic expression. In extenuation may we plead that we got our information on the Coca-Cola dispenser from the artist who designed it, and he said he used Bakelite.

The designers did put on a good show. Whether the "right people" were attracted we don't know, but on the day we were there some 1,100 of the common run paid 35 cents apiece to look at the show.

Horsepower Charts

PRODUCT ENGINEERING
NEW YORK

Please let us know if you can furnish a copy of the June, 1933, issue of *Product Engineering*. In this issue, on pages 239 and 240, you publish six tables on the horsepower capacity of straight bevel gears together with an explanation of the use of these tables, and in a later issue you published the remaining seven tables.

We would like to know the cost of both the June, 1933, issue and the later issue which contained the balance of the tables.

C. J. LEIFELD, *Purchasing Agent*,
The Jeffrey Manufacturing Co.

ALTHOUGH complete copies of the issues of *Product Engineering* happened to be available we sent Mr. Leifeld clippings of the horsepower tables in which his company was interested. By so doing we saved him all expense because clippings cost nothing. And that always appeals to a good purchasing agent. But at the same time we saved ourselves a draft on our very small supply of back numbers. We mention this to remind those who want file copies of specific articles or departments that it is not necessary to request and pay for complete copies of specific issues. We can usually supply a clipping and we are glad to do it without charge.

Motor Inquiry Form

PRODUCT ENGINEERING
NEW YORK

The supply of this form that you sent us was all used up in distributing them to our District Offices, and we have had a number of favorable comments on this form.

One office in particular would like to have an additional supply, and we believe other offices will come through with similar requests. If it is possible, therefore, we would appreciate your sending us another batch of this form. Perhaps you could spare 100 or 150 more copies. Thanks very much.

The Ideal Electric Manufacturing Company,
Synchronous Division

WE were only too pleased to send 150 copies of this form which was published in the April number of *Product Engineering* and which has been so popular. The funny thing about it is the reluctance of readers and advertisers to ask for the quantities they can really use. Don't be bashful. We prepared this form as a service to our field, we put our name on it, and we expect to be repaid many times over for the small expense involved by the advertising and good will which its distribution will bring us.

Here are a few samples of the very modest requests for supplies of it from other readers:

The inquiry form for electric motors is very nicely worked out and we could use a number of these to good advantage. However, we would not feel like asking you to supply them without cost and we suggest that you quote us a price for a lot of 100.

. . .

We will be very glad to receive one-half dozen copies of inquiry form for electric motors.

. . .

This form was discussed with several members of our Sales Department and they believe it so useful that I have been requested to obtain a supply. If you will send us about 150 additional copies we can assure you that they will be put to good advantage.

. . .

Our electrical division would like to have 500 copies of this inquiry form and I am wondering if you have these available. If you do not have them printed, will you give me an idea what the cost would be of running off 500 copies?

. . .

If it will be agreeable to you, we would like to have about 100 or 150 copies, and if you are in a position to furnish these in still greater quantities, you might let us know what the charge might be in say lots of 2,000, 3,000 and 5,000.

. . .

We would be pleased to receive from you one dozen copies of inquiry form for electric motors.

For Translation

PRODUCT ENGINEERING
NEW YORK

We refer to the March issue of *Product Engineering* containing a case history of our type "BB" burner.

I am very much interested in obtaining three additional copies to be sent to our Paris Office for translation and distribution to our European dealers.

I will be very grateful for your prompt compliance with our request.

S. S. STROUSE, *Export Manager*,
May Oil Burner Corporation

THE three copies went forward some days ago, and by this time the Communists of Paris have probably stopped fighting the police to read all about modern American oil burners. If they are not doing so may we respectfully suggest that such an occupation would be much more profitable than rioting?

To Moscow

PRODUCT ENGINEERING
NEW YORK

Kindly help me to obtain the trade publications mentioned in *Product Engineering*. I inclose clippings of the items telling about the ones I should like to have.

M. L. SOROKIN,
Moscow

THE Soviets are progressing. Mr. Sorokin's desire for educational publications includes everything from Norman Bel Geddes' book on streamlining to the latest pamphlets on anti-friction bearings.

PRODUCT ENGINEERING

KENNETH H. CONDIT, *Editor*

MCGRAW-HILL PUBLISHING CO., INC. JUNE 1934 VOLUME FIVE, NUMBER SIX

You Can't Stop Progress

IN THE course of an inspiring speech at the recent meeting of the Chamber of Commerce of the United States, Charles F. Kettering had this to say: "This civilization of ours is an experiment. It is the first time anybody ever tried to do things like this; it is the first time we ever had a lot of automobiles, a lot of radios, a lot of telephones, and a lot of locomotives. We make a mistake when we monkey with these new things. Let's not go and throw them all away and say we are not going to have any more new things, because you can't keep from having new things, because you just don't have anything to say about it any more than you have when it is going to rain and when it is going to thunder, or anything else."

Mr. Kettering is right. You can't prevent the ingenious American from inventing new ways of doing things, and new devices with which to do them. It is his nature to act that way and even depression does not stop him for long.

Company executives of the reactionary type, the ones the President refers to as "Tories," may lose faith in the future of the country and buy farms to which to retire when everything goes to pot. They may clamp down on any plans for expansion, or for replacement of worn out or obsolete machinery, or for engineering development and research work looking to improvement of their products. But they will not stop the progress of the country or the industry, for new companies headed by younger more aggressive men will press forward, experimenting with new products and developing them commercially, and will eventually displace the reactionary organizations that do not mend their ways.

By the same token, those members of the "Brains Trust" who are endeavoring to stimulate recovery by restricting the use of machinery and by retarding the march of invention will be equally unsuccessful. Theirs is an effort to apply the remedies of an economy of scarcity, such as exists in Russia, to an economy of plenty, which has been our lot in this country for some years. It is failing just as every other attempt to stop progress has failed.

Support the Move for A Design Division

WE HAVE heard complaints that none of the major engineering societies properly recognize the field of product design. That there is some basis for such complaint is indicated by an analysis of 150 papers presented at the annual meetings of three leading engineering societies last winter. Only 19 of these papers dealt exclusively with mechanical design.

Society members through their committees, divisions and elected officers control the activities of the organization. If the conditions do not satisfy designers, it is their responsibility as members to undertake corrective measures.

A group in the American Society of Mechanical Engineers has been sponsoring a movement for more society activities from which the designing engineer might receive benefit. This move received the active support of officers of the A.S.M.E. and the immediate result was the organization of a machine design committee in the Machine Shop Practice Division.

If enough members of the A.S.M.E. evidence a genuine interest in having a Product Design Division, their desires will undoubtedly be met. By making known their interests, by submitting papers and by taking an active part in meetings devoted to product design subjects, members of the A.S.M.E. can convince the officers of the society of the need for such a division.

Let us hear about this from interested engineers.

Bronze Bearing Alloys
Their Properties and Applications

Bearing alloy properties required to meet various conditions of speed, lubrication, load and the chemical composition and physical properties of ten bronze alloys for different fields of application

C. H. LEIS
Chief Engineer, Johnson Bronze Company

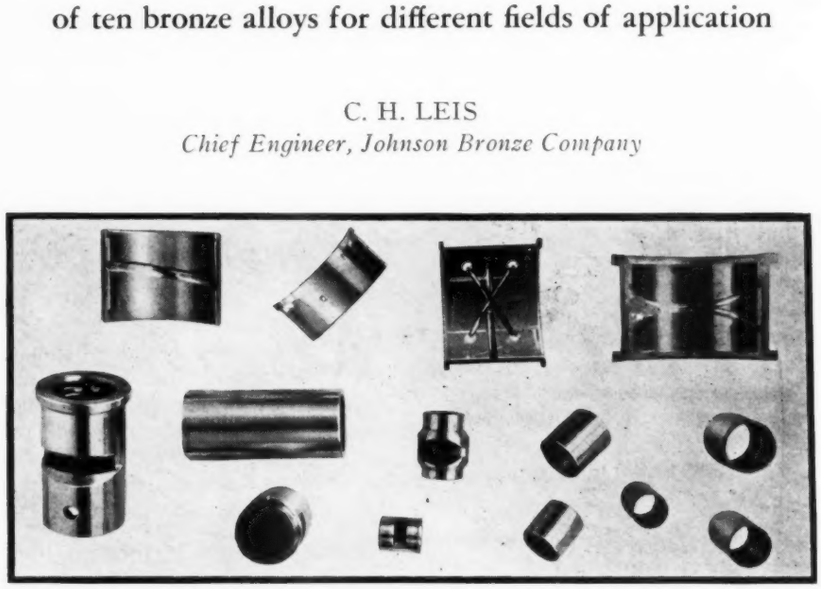


Fig. 1—A group of plain cast, rolled sheet and graphited bronze bearings

DIVERSE applications of cylindrical type bearings make necessary the availability of a variety of bearing metals possessing many distinctly different properties. Each particular bearing metal will be best suited for those conditions of service wherein its outstanding characteristics will be of greatest advantage. Accordingly, in the selection of a bearing material it is necessary that the service conditions and the destructive forces on the bearing be analyzed first and the bearing material be then selected for that specific application. In designing the bearing, further consideration must be given to speed, load and other operating conditions.

To definitely establish the significance of the foregoing, let us consider the conditions that establish the relative importance of the physical properties that determine suitability of a bearing metal. In any bearing, when the shaft is at rest its surface is in direct contact with that of the bearing. On starting, the rotation of the shaft carries the lubricant around the bearing and if the combination of operating conditions is correct, sufficient pressure will be built up in the film of lubricant to separate the bearing and the shaft, as illustrated in Fig. 2. But regardless of how large the bearing may be or how efficient the lubricating system, every time the shaft is stopped it makes surface to surface contact with the bearing.

Metal to metal contact between shaft and bearing will exist until the shaft speed becomes sufficient to build up the necessary pressure in the film of lubricant. In every bearing installation the ability of the bearing to resist wear during this period of metal to metal contact without damage to either the bearing or the shaft, is of utmost importance. This is well illustrated in Fig. 3 which pictures the results of a test. Two different alloys were subjected to exactly the same test conditions. The bearing at the left, having poor wearing characteristics and lacking plasticity, developed a rough surface which scored the shaft and rendered both shaft and bearing unfit for further service. In the bearing at the right, the metal had better wearing characteristics and more plasticity, and as a result, both shaft and bearing are in a sufficiently good condition to render further service.

The fact that all bearings must operate for some part

of their life in contact with the shaft, necessitates a careful selection of a suitable bearing material for a given operating condition.

The following physical properties govern the suitability of a bearing metal: (1) Plasticity; (2) Ability to Resist Wear; (3) Coefficient of Friction; (4) Compressive Strength; (5) Resistance to Pounding; (6) Toughness.

Plasticity

A bearing material should have the ability to conform itself to the shaft. The ability to do so is termed plasticity. This characteristic permits deformation under compression to compensate for misalignment and microscopic irregularities on the bearing surfaces. The degree of plasticity is not only different for various bearing materials but for any given material it increases with temperature. Thus, the heat generated on a high-speed shaft with a light load may be equal to that generated in a slow-running shaft with a heavy load. Obviously the bearing material for the former must be sufficiently plastic to conform itself under light loads while the latter condition calls for a material which must be less plastic in order to have sufficient load carrying capacity.

Ability to Resist Wear

The physical ability to resist the destruction of two rubbing surfaces without lubrication is termed wear resistance. Alloys listed herein have their wear-rate tabulated in grams per 10,000 revolutions, as established on the Amsler wear-test machine. In Fig. 5 are given curves showing the results of such wear tests conducted by the Bureau of Standards, whose *Research Paper No. 68* gives a detailed account of these tests.

Coefficient of Friction

The coefficient of friction is the ratio between the force of friction and the normal pressure, expressed as a decimal. Under given conditions the heat generated in a bearing operating in contact with a shaft is proportional to the frictional resistance (the coefficient of friction) between the bearing and the shaft material. Should the coefficient of friction be high, the heat generated even during short periods of metal to metal operation might be sufficient to do considerable damage, either to the bearing or other adjacent parts. Naturally the lower the coefficient of friction, the less will be the chance of damage through heat of friction.

In the Bureau of Standards *Research Paper No. 205* is given the results of tests on the coefficient of friction of copper-tin-lead alloys as determined on the Amsler wear-test machine. The accompanying Fig. 4 is reproduced from that report.

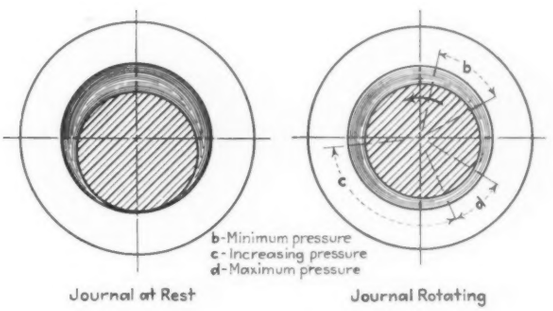


Fig. 2—Bearing metals must withstand the wear of the metal-to-metal contact when rotation begins

Compressive Strength

A necessary requisite of a good bearing metal is that it have sufficient compressive strength to support the

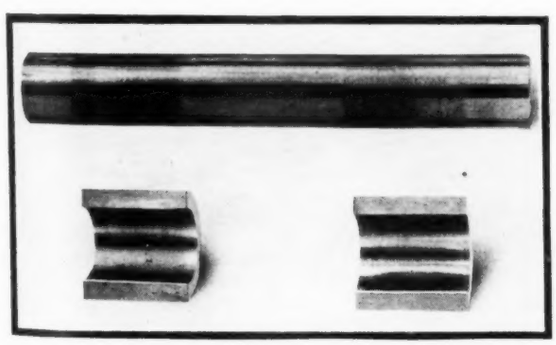


Fig. 3—Poor wearing characteristics and lack of plasticity caused failure of the bearing at the left whereas the bearing at the right is still in good condition

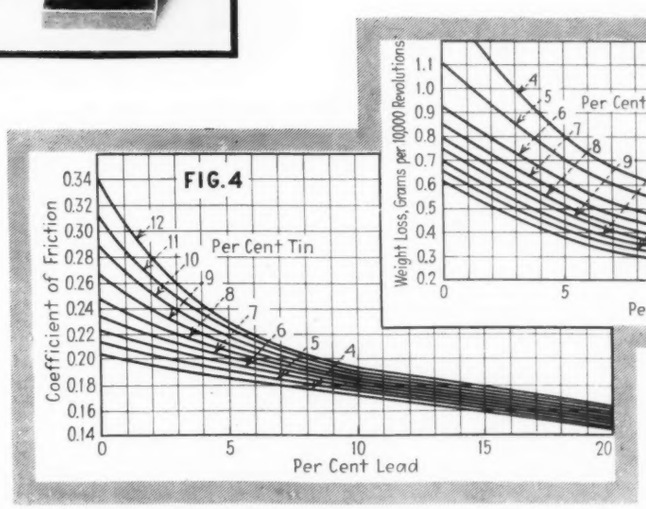


Fig. 4—Coefficient of friction for copper-tin-lead alloys, plotted from the results of tests on the Amsler wear-test machine

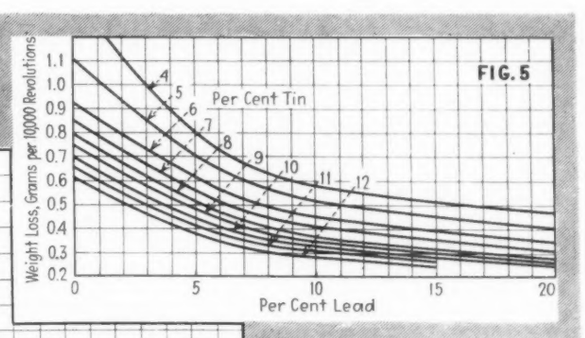


Fig. 5—Wear rate of copper-tin-lead alloys as determined by tests conducted by the Bureau of Standards on the Amsler wear-test machine

loads imposed upon it. However, the load carrying capacity of a bearing is often limited not by the bearing material but by the inability to maintain a lubricating film between the two surfaces.

The tensile strength, yield point and proportional limit of copper-tin-lead alloys is shown by the curves in Fig. 6. In Fig. 7 is shown the per cent elongation and reduction of area.

Resistance to Pounding and Toughness

Resistance to pounding is that quality which permits impact loads without permanent deformation. When impact loads occur, the oil film may be ruptured. To meet this condition the bearing material must have a low coefficient of friction, a high wear rate and must not deform when subjected to pounding. Further reference to this will be made later. Comparative resistance of various copper-tin-lead alloys to repeated pounding is given by the curves in Fig. 8, which are based on the results of tests conducted by the Bureau of Standards. The curves in Fig. 9 give the Brinell hardness.

For constant and continuous motion where impact loads do not occur, toughness (resistance to fatigue) is of little value in bearing applications. In Fig. 10 are given the curves showing the results of tests conducted by the Bureau of Standards.

Plain Cast Bronze

Bronze is a copper-base alloy. Tin is a predominating alloying element. Originally copper-base bearing metals in common use were copper-tin alloys to which a small amount of zinc was added as a deoxidizer. In later years, however, the percentage of copper-tin used in bearings has been decreasing rapidly and instead, a relatively new type of bearing alloy has taken its place. These newer alloys are, in effect, bronzes to which varying amounts of lead have been added. These are called copper-tin-lead alloys.

The superior wearing qualities of these metals are attributed to the peculiarities of their structure. Bearing bronze is an alloy, with an heterogeneous arrangement of the elements, in which the copper-tin composition forms a matrix to hold the softer plastic materials. Thus the harder elements carry the load while the softer and more plastic metal allows conformability to the shaft. The proportion of the copper-tin-lead alloy can be varied to suit almost any operating condition, and it is important that the alloy be selected to meet the conditions under which it must perform.

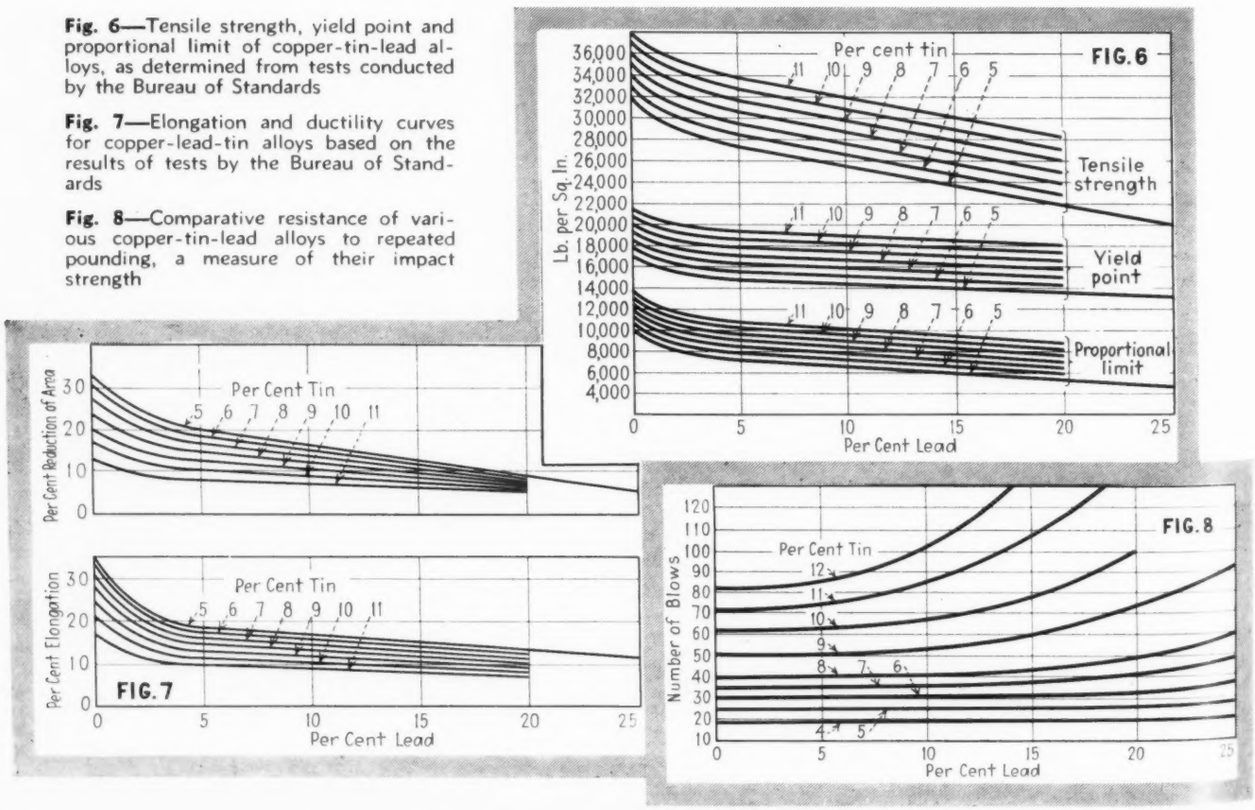
In general the composition of copper-tin-lead bearing alloys is within the ranges of 5 to 11 per cent tin and 0 to 25 per cent lead, the balance being copper. The poor resistance to pounding of some alloys outside this range, in particular copper-lead alloys of the composition of about 70 per cent copper and 30 per cent lead, has so restricted their use that they do not warrant consideration as cast bearing bronzes.

On page 240 (Reference Book Sheet) is given a chart wherein ten alloys, whose physical properties and chemical composition are tabulated on the page preceding, are grouped according to the field of service for which they are best adapted. The use of this table and chart will be found helpful in selecting the proper alloy to meet specific operating conditions.

Fig. 6—Tensile strength, yield point and proportional limit of copper-tin-lead alloys, as determined from tests conducted by the Bureau of Standards

Fig. 7—Elongation and ductility curves for copper-lead-tin alloys based on the results of tests by the Bureau of Standards

Fig. 8—Comparative resistance of various copper-tin-lead alloys to repeated pounding, a measure of their impact strength



Despite the fact that the properties of copper-tin-lead alloys are relatively insensitive to minor changes in either tin or lead content, there are a large number of specifications in use today. These specifications often differ in tin or lead content only to a minor extent from standard alloys, and their performance could be no different than that of a standard alloy. Special alloys are somewhat more expensive, unless the quantity desired carries sufficient tonnage, and cannot be obtainable as quickly. The preferred list of alloys presented here cover all usual requirements.

In giving physical properties we have allowed for a safety factor so that engineers will be perfectly safe in using these particular figures in any of their applications.

In addition to the chart and table of properties (R.B.S. pp. 239-240), the following brief descriptions point out the dominating characteristics of the bearing metals listed in the table and give their fields of general application. For identification purposes, the alloys are designated according to the alloy numbers as given them by the Johnson Bronze Company. Wherever the alloys correspond to the specifications of organized societies or government bureaus, the corresponding specification numbers are given.

Fig. 9—Brinell hardness of copper-tin-lead alloys containing 5 to 11 per cent tin, as determined by tests conducted by the Bureau of Standards

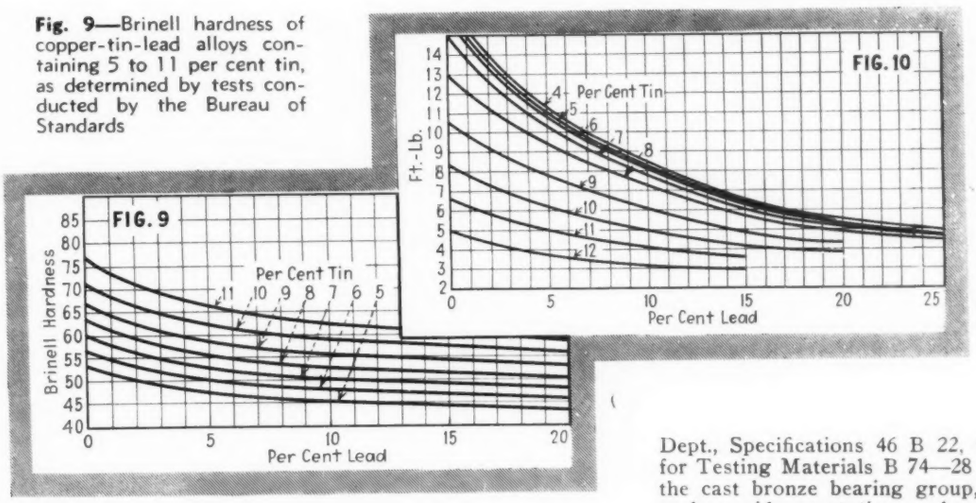


Fig. 10—Izod notch-toughness of copper-tin-lead alloys containing 4 to 12 per cent tin, based on the results of tests conducted by the Bureau of Standards

Sleeve-type bearings that will be described in these articles include plain cast bronze, rolled sheet bronze, laminated bearings and graphited bronze bearings. These are listed here in the order of their wide useage.

JOHNSON ALLOY NO. 19 has an advantageous tin-copper structure which is required in a good bearing bronze. The bearing crystals are amply surrounded by lead, giving the structure exceptional bearing qualities.

Where lubrication is hard to maintain, because of inaccessibility, on account of abnormal temperature or through other causes, this alloy will give good results. Its high ratio of tin-copper, makes it slightly brittle and it does not possess the toughness found in some of the lower tin ratio alloys. Although it has a high resistance to pounding it should only be used when the outer surface of the bearing is backed up rigidly by the housing.

Principal Applications: This alloy is well adapted for mill bearings, general machinery bearings, excavating machinery bearings, gas and diesel engine bearings, impact and pulverizing machinery bearings, textile machinery.

JOHNSON ALLOY NO. 25 (Plastic Bronze) has a low tin-copper ratio, therefore its compressive strength is very low. Under heavy loads it crushes and deforms more readily than do alloys which possess a more evenly balanced tin-copper ratio. It has a very low coefficient of friction, an unusually low resistance to pounding, and conforms itself readily to the shaft.

Principal Applications: This alloy is well adapted for high speeds with light to medium loads, and for applications generally free from shock. In pump bearings and cylinder sleeves it has proven itself valuable, being a good acid resisting alloy. It is used extensively in small electric motors, electric fans, conveyors, lapping machines, sanding and woodworking machinery bearings and for bottling, labeling and wrapping machinery, and where lubricaiton is not dependable.

JOHNSON ALLOY NO. 27 (S.A.E. No. 64; U.S. Army Ordnance Dept., Specifications 57-70B, Grade 8; U.S. Navy Dept., Specifications 46 B 22, Grade 2; American Society for Testing Materials B 74—28 T; A.R.A.—Grade "C"). In the cast bronze bearing group, no other alloy has earned such a wide reputation as has alloy S.A.E. No. 64. The broad range of applications has caused its acceptance by engineers who specify this analysis quite universally because of its fine performance under heavy loads. This alloy has an exceptionally low coefficient of friction. Its load carrying capacity is high and in addition the copper-tin constituents are well surrounded with lead, resulting in a superior general purpose bearing material. Alloy No. 27 has the following important physical characteristics: low coefficient of friction, high resistance to pounding, conformability and exceptional load carrying capacity. The suitability of this alloy is outstanding under severe operating conditions. It can be used with hard or soft shafts. At elevated temperatures, it will retain its strength in much better proportion than most other bearing alloys.

Principal Applications: The principal applications of this most widely used bearing material include: locomotive crossheads, crankpin bearings, lathe and grinder bearings, large motor bearings, general automotive bushings and roll neck bearings. For all general machinery bearings, where speeds are high, favorable results may be obtained since it is an outstanding general purpose bearing bronze.

JOHNSON ALLOY NO. 29, (S.A.E. No. 67 U.S. Army Ordnance Dept. Specification No. 57-70B, Grade 7; U.S. Navy Dept. Specification No. 46 B 22, Grade 1; Federal Specification Board No. 172 A, Grade 7; General Motors Company, Specification No. G.M.C. 4073-M). This is a high lead alloy and is valuable as an acid resisting bearing bronze. Because of its higher tin-copper ratio it can be used with higher loads than Alloy No. 25. Compared with Alloy No. 27 (S.A.E. No. 64) however, it has less load carrying capacity and impact resistance, because of its higher lead content. Its plasticity is particularly valuable

when used with soft shafts operating at high speeds, because it conforms itself readily under comparatively light loads, compensating for slight misalignments and microscopic irregularities on the bearing surfaces.

Principal Applications: This alloy is recommended for bearing use in acid pumps, mining machinery, small light rolling mills, knitting machinery and fractional horsepower electric motors.

JOHNSON ALLOY NO. 51 is a leaded gun metal. The 1 per cent lead in addition to improving the bearing and machining qualities, reduces the susceptability to seizure which is often encountered with gun metal. As a bearing material it requires dependable lubrication, because of its high coefficient of friction and unusually high wear rate. It withstands severe shock and abrasion, but lacks plasticity and does not conform under light loads. Where torsional strength is required in addition to a bearing surface, this alloy qualifies.

Principal Applications: Alloy No. 51 is good for bearings subjected to abrasives, dust and grit, as in coal crushing, grinding and pulverizing machinery, power shovels, concrete mixers and ditch digging equipment. It is used also in slow-speed diesel engines for connecting rod bearings, automotive rocker arms, piston pin bushings and in worm gears.

JOHNSON ALLOY NO. 53 (S.A.E. No. 63; U.S. Army Ordnance Dept. Specification No. 57-70B, Grades 5 and 6; U.S. Navy Dept. Composition "G", Specification Nos. 46 B 5 f and 46 M 6e; Federal Specification Board 172 A. Grades 5 and 6; General Motors Company, Specification No. G.M.C. 4048-M; American Society for Testing Materials B 10-18) contains a high ratio of tin-copper structure, but is not interwoven with soft lead, therefore lacks plasticity. It is a tough bronze good for heavy duty service, low rubbing speed, where abrasive action takes place, and where vibration and shock are encountered. A hardened or semi-hardened steel shaft should be used with this alloy. With insufficient lubrication it will gall because the high copper content tends to plate it to the shaft and grab. It can be used in places where considerable dust accumulates, as it will withstand the abrasive action better than most bearing metals.

Principal Applications: This alloy is used successfully in both gears and gear bushings, for bridge bearings and wearing plates, for piston pin and valve guide bushings, and in general where heavy duty bearings are needed and heavy pressures occur. It withstands heavy shock and impact loads well but flooded lubrication is imperative to secure the best results.

JOHNSON ALLOY NO. 55 is one of the hardest standard alloys used in bearing applications and should be used only with hardened steel shafts. Its hardness is by virtue of its high ratio of tin-copper structure. With sufficient lubrication, it withstands heavy loads and shock. This alloy resists abrasion and its use is restricted to the above operating conditions.

Principal Applications: Generally used on heavy machinery in pulp and paper mills where contacts with sulphites are prevalent; in mine pumps in contact with sulphurous acids and where abrasive conditions prevail, as in concrete and road machinery, in conveyors for coal, sand, rock and cement. Also used as piston pin bushings for trucks and tractors. In some instances it will serve for use in valve guides.

JOHNSON ALLOY NO. 66 is extremely soft and might be termed a semi-plastic bronze. The ratio of tin to copper is extremely low and compared to the plastic bronzes the lead content is low. Because of its low tin content it is inexpensive. With good lubrication, this bearing material may be used where the loads are light and speeds are medium-high. Soft or hard shafts can be used with this alloy. It has such a low pounding resistance that it should never be considered where impact loads occur.

Principal Applications: For economic reasons it may be used in fractional horsepower motors and in light duty machines where no shock loads are encountered, as in bench lathes, small wood-turning machines and similar equipment.

JOHNSON ALLOY NO. 71, more commonly known as "Steam-Metal" has a fairly high compressive strength and its rigidity is excellent. This alloy is easily cast and its machinability is good but since the material is somewhat soft it will not withstand continual hammering. As a bearing alloy it can be rated only fair and its application should be selected carefully. It will wear rapidly and offers only a low resistance to pounding.

Principal Applications: Alloy No. 71 rates well for use in stuffing box glands and cylinder sleeves and in valves for steam or liquid. Its use in bearings is confined principally to the backing metal in bronze-backed babbitt lined bearings. For the manufacture of parts requiring threading operations it will be found satisfactory. It is also used for sliding valve seats.

JOHNSON ALLOY NO. 72 has a low ratio of tin-copper structure surrounded by small quantities of lead. Its compressive strength permits its use in applications where shock or impact loads are light. It has good bearing qualities when used with soft shafts for medium loads and speeds, and for vibratory loads such as encountered in fractional horse-power motors. In comparison to Alloy No. 27 its wear rate is slightly higher, its coefficient of friction is the same, but its resistance to pounding is much lower. Its application to general uses is not as broad as Alloy No. 27, and its low resistance to pounding limits its field to carefully selected applications.

Principal Applications: This alloy is used in cam-shaft bearings, gear bushings, loose pulleys, fractional horsepower motors, in small starter and generator bushings, automotive spring bushings, and in general in light machinery where only medium loads are encountered and no impacts or shocks occur.

Editor's Note—The next article will cover sheet bronze bushings, laminated bearings and graphited bushings of the cast and sheet bronze types.

Internal Stresses in Welds

HIGH tensile stresses parallel to the welded joint are almost always near the yield point in both electric and gas welds according to measurements recently reported in *Transactions* of the American Society of Mechanical Engineers, April, 1934, page 249. In most cases stresses parallel to the seam are higher than at right angles.

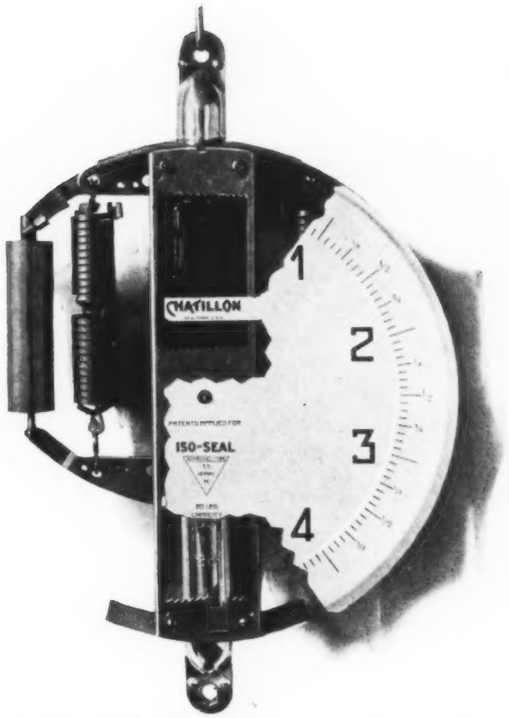
High initial stresses under static loads are not necessarily dangerous in a ductile material, since local creep distributes internal stresses when the yield point is exceeded. Under repeated loading the stress distribution is also leveled if the material is ductile, although the initial stresses may reduce the factor of safety. But under impact loads, the presence of initial stresses is said to be extremely undesirable.

Test methods used in this study are based on a disturbance of the equilibrium of forces. A small hole is drilled in the stressed portion and the small deformations caused by the hole are measured. Since the hole is so small that it does not destroy the part, the method is shown to be applicable to castings, rolled structural shapes, and finished structures, as well as welded parts.

Scale Springs

99.92 Per Cent Accurate

Metallurgical development and mechanical design eliminate errors arising from the four variables inherent in commonly used spring materials



Self-compensating springs in the Chatillon scale are made from a cold-worked, 36 per cent nickel-chrome steel. Two springs, oppositely wound, are connected in series to prevent twist

DEVIATION from true proportionality between load and deflection in springs for scales, testing machines and instruments must be compensated if errors are to be less than 1 per cent. Such deviations, present in all common spring materials, are caused by: (1) A change in elastic modulus with changes in temperature, (2) Non-proportionality between stress and strain, (3) Creep under constant load or a lag of strain in relation to stress, which is present even under conservative working stresses, (4) Back-error or hysteresis, analogous to backlash in gears, step-by-step-loading and unloading curves producing a closed loop instead of a single stress-strain curve.

In a recent development combining the efforts of the research staff of the John Chatillon & Sons, Prof. M. F. Sayre of Union College, and A. V. deForest, consulting metallurgist, a new type of helical spring has been designed in which each of the deviations is inherently compensated within a total error of 0.08 per cent.

New Spring Material Developed

Temperature errors are eliminated by the use of a special alloy developed by Mr. deForest. This alloy, of the Elinvar type, contains 36 per cent nickel, 8 per cent chromium, a total of 4 per cent of other elements including manganese, silicon, molybdenum, copper and vanadium, and the remainder iron. About 90 per cent reduction in area by cold drawing increases the safe working stress from 12,000 to 60,000 lb. per sq. in., and a moderate temperature heat-treatment stabilizes the spring, reducing creep error to less than 0.02 per cent and back error to less than 0.04 per cent.

Errors of non-proportionality are brought under control by a mathematical analysis developed by Professor Sayre. Straight-line deviation in helical springs is shown to be a product of three factors: the load and the original dimensions of the spring, the proportion between the stiffness in bending and the stiffness in torsion of the wire in the spring, and the spring extension, as given by the following equation:

$$Deviation = \frac{PR_0^2}{3GJL}\left(3 - 2\frac{GJ}{EI}\right)(H^2 + HH_0 - 2H_0^2)$$

In this equation P equals load on spring, R_0 equals initial radius of coil, E equals tension modulus, G equals torsion modulus, I equals moment of inertia in bending, J equals stiffness factor of wire in torsion (polar moment of inertia for round wire), L equals length of wire in spring, H equals total height and H_0 equals initial height of effective portion of spring.

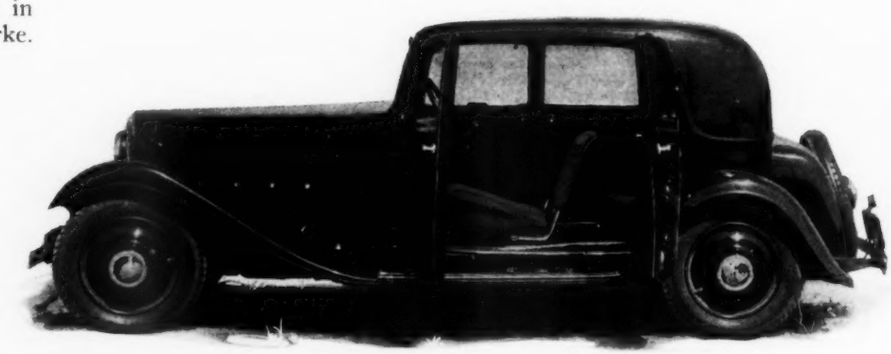
The first and third factors can be varied by changing the general design of the spring but neither one can be reduced to zero. By properly choosing the shape of the wire in relation to the elastic constants of the material, however, the factor GJ/EI can be made to equal 1.5 and the theoretical deviations from straight-line behavior is equal to zero. Using a wire with a rectangluar cross section these helical springs are said to have non-proportional deviations of less than 0.01 percent.

As a result of this development, the maximum error from temperature variations, non-proportionality between load and deflection, back-error and creep in the spring system shown in the illustration is said to be not more than 0.08 per cent.

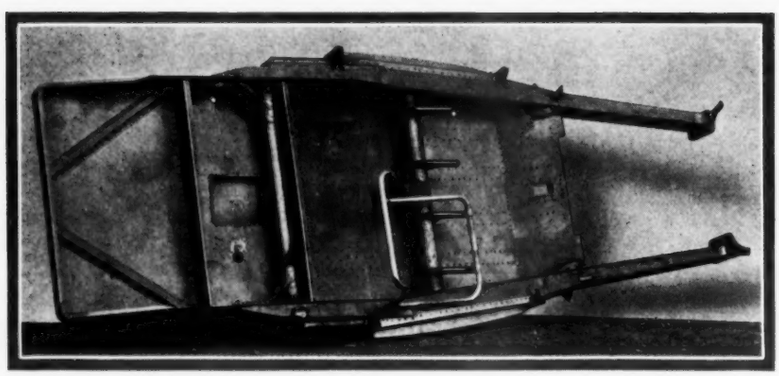
Developments In PRODUCT DESIGN

• German Developments in Auto Bodies

A type of all-steel automobile body construction entirely foreign to American practice is used in the 4-cylinder engine, front drive Adler-Trumpf, pillarless four-door sedan built by the Adler-Werke, Germany. Independent springing is used on all four wheels. The all-steel body and chassis unit is manufactured by the Ambi-Budd Presswerke associates of the Edward G. Budd Manufacturing Company, and was developed by the Ambi-Budd Presswerke in cooperation with the Adler-Werke.



Pillarless four-door sedan with all-steel body-and-chassis unit developed by the Ambi-Budd Presswerke for Adler-Trumpf. The chassis and body forms a welded structural unit with tubular cross members for stiffeners



As shown by the illustrations, the chassis is substantially the floor of the body. The finished body and chassis form a single structural unit, all parts being welded together.

With reference to the floor of the body, which serves as a chassis, in the accompanying picture of it can be seen the brackets at the front end, to which are attached the arms for the independently sprung wheels. Directly under the front end of the toeboard is a tubular transverse member which also supports the engine block. The tubular cross-member in the middle also serves as a support for the adjustable front seats and as a heating duct. The rear tubular cross-member supports the independently sprung rear axles while the shock absorbers are attached to the diagonal braces. The depression in the floor just to the rear of the rear tubular cross-member is for holding the battery.

One immediate result of this type of construction has been the elimination of more than 200 lb. dead weight. And in place of the conventional construction wherein a whole series of related members are attached to one another by mechanical fastenings and the chassis and body are separate, this structure presents a rigid single unit steel structure. And by reinforcing the side members at the floor of the body, it has been possible to entirely eliminate the usual pillars, as can be seen in the picture of the finished car.

Other features of the car include light-weight pistons, three main bearings for crankshaft, battery ignition, pressure lubrication, four speeds forward and one reverse, and swinging-arm type rear axle pivoting on rubber set bolts.

• Stainless Ice-Cube Trays

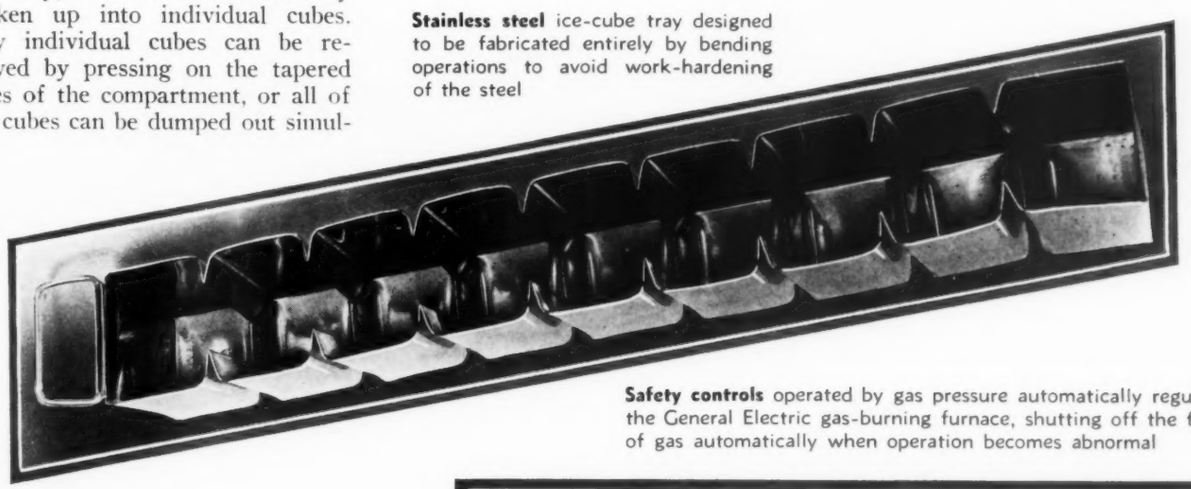
Because the material is work hardening, it is usually necessary to anneal stainless steel sheets if they are to be given successive cold-drawing operations. These difficulties and the expense involved prevented the deep-drawing of stainless steel for ice cube trays used in domestic refrigerators. However, the McCord Radiator & Manu-

facturing Company ingeniously circumvented all of these difficulties by a design of ice-cube trays made of stainless steel sheets, without the use of drawing operations. In their Metlflex ice tray, shown in the accompanying illustration, the finished product is made from a flat sheet solely by bending or creasing operations.

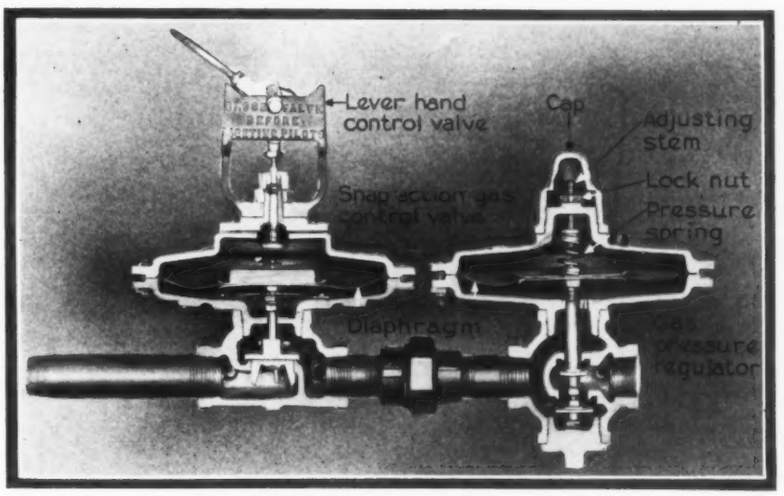
One of the features of this tray is its flexibility. By simply flexing the tray, the ice is immediately broken up into individual cubes. Any individual cubes can be removed by pressing on the tapered sides of the compartment, or all of the cubes can be dumped out simultaneously. And because all of the sides of each compartment are tapered, it is not necessary to first free the ice by thawing.

Other advantages claimed for these stainless steel ice-cube trays are a greater rate of heat conductivity, sanitary surfaces and nonrusting.

Stainless steel ice-cube tray designed to be fabricated entirely by bending operations to avoid work-hardening of the steel



Safety controls operated by gas pressure automatically regulate the General Electric gas-burning furnace, shutting off the flow of gas automatically when operation becomes abnormal



• Pressure Operated Valves for Gas Furnaces

Operated by a snap-action diaphragm, the valve supplying gas to the burners in the new General Electric gas furnace is actuated by gas pressure from the mains, rather than directly by electrical controls. Gas pressure normally admitted to the lower face of the snap-action diaphragm holds the valve open, except when pressure is also admitted to the upper face of the diaphragm. With the same pressure on both faces of the diaphragm, an adjustable weight carried by the snap-action diaphragm, snaps the valve shut, cutting off the supply of gas to the main burners.

Mounted alongside the water-level gage is a solenoid valve controlled by the electrical circuit of the room thermostat. When heat is needed, the solenoid valve cuts off gas pressure to the upper side of the diaphragm, venting the gas through an escapement pilot burner. The pressure below the diaphragm then opens the gas valve. In shutting off the heat, the solenoid valve admits gas pressure to the upper chamber of the diaphragm, snapping the gas valve shut.

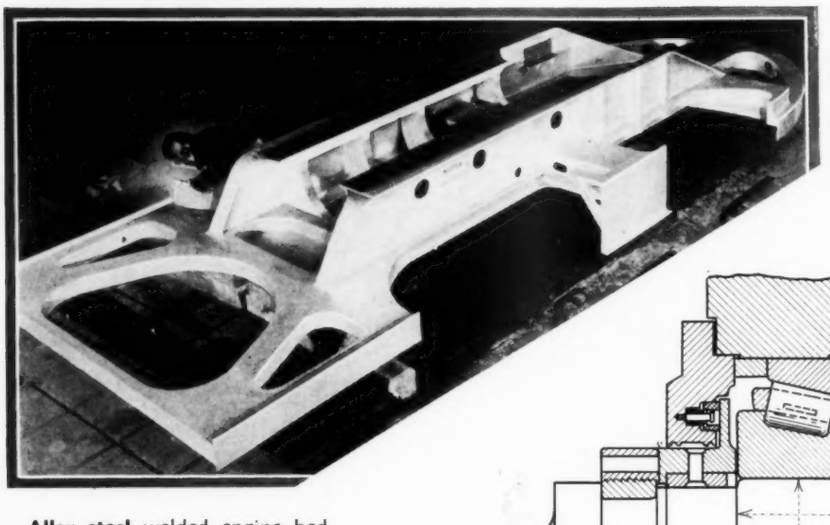
Superimposed on the thermostatic control, a safety low-water and excess-pressure control by-passes the solenoid valve, admitting gas pressure to the top of the diaphragm, and shutting down the furnace. Thermostatic elements over the pilot lights actuate bleed valves that allow gas pressure to build up on the lower side of the snap diaphragm only when the pilots are burning. Thus, the gas valves remain closed when the pilots are out, when the pressure is too high, when the water is low, or when for any reason the main supply fails.

Secondary air-control valves, positively operated by gas pressure, are used in addition to the gas control system to admit air to the furnace when gas is supplied, and to shut off the secondary air when the main flames are extinguished. By preventing air from sweeping through the boiler, losses are reduced.

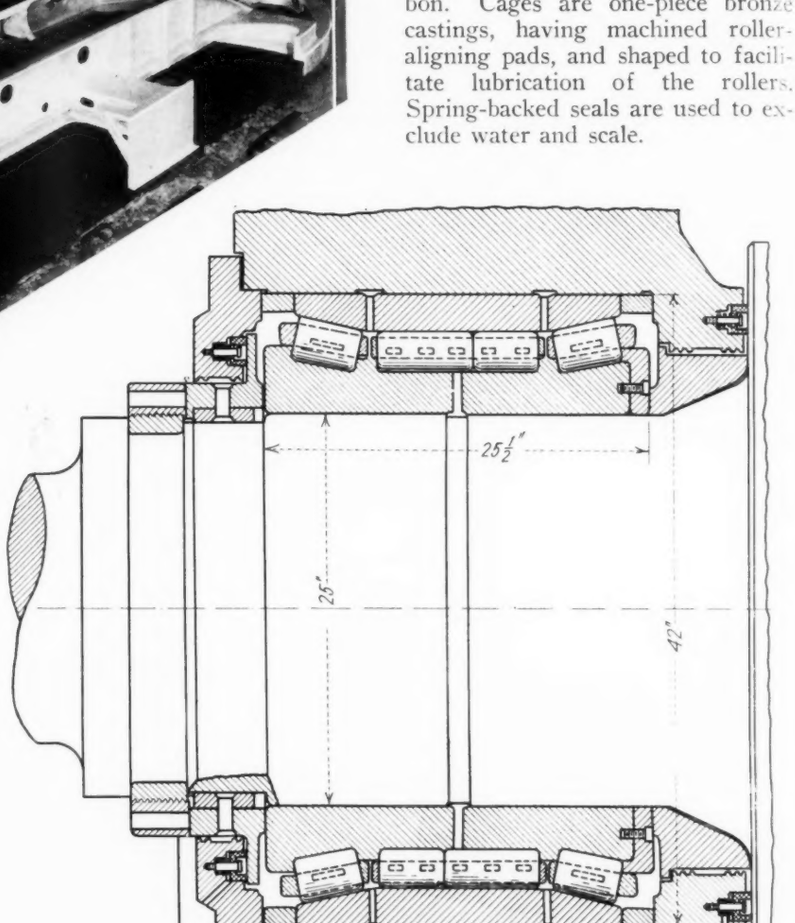
• Welded Alloy Steel Engine Bed

To meet the requirements of light-weight, strength and rigidity, an unusual design of engine bed was developed for the Burlington streamlined train. This engine bed combines the front bumper, the engine bed and motor bed in a single arc welded structure, 25 ft. 3½ in. long, by 8 ft. 8 in. wide.

Fabricated in the weldery of Lukenweld, Inc., 204 separate steel plates cut to shape by oxy-acetylene



Alloy steel welded engine bed for the Burlington streamlined train. This unit structure serves as bumper, engine bed and bolster and weighs only three tons



Combined radial and thrust bearing for roll-neck journals, developed by Bantam Ball Bearing Company for a four-high cold reduction mill built by E. W. Bliss Company, representing the largest of such bearings ever used

flame and formed as required, were welded together to form the completed structure which weighs only three tons. After being welded, the bed was annealed to relieve any possible internal stresses.

The steel used for this bed is a Cr-Mn-Si high-strength alloy known under the tradename of Cromansil, having a tensile strength of 90,000 lb. per sq. in., and an elastic limit of 70,000 lb. per sq. in. This material also possesses an unusually high reistance to fatigue and shock.

• Roller Bearings for Rolling Mill Service

Extreme severity of service in roll-neck bearings for steel mills makes the design of anti-friction bearings for this application a difficult problem. Thrust loads are present in addition to large radial loads, good alignment is seldom obtained, the bearings must be readily removed for changing rolls, and space is restricted by short center distances between rolls.

Because of the load conditions the results obtained with anti-friction bearings depends greatly on the load distributing characteristics of the bearing itself. In a roller bearing used in a mill, made by the United Engineering & Foundry Company, the bearing is designed to follow load deflections, distributing the bearing load over the entire length of the bearing.

In the roll-neck bearing shown in the illustration, both straight and tapered rollers are used, a row of tapered rollers being used at each end. By the use of short rollers, smaller roller diameters can be selected, reducing the outside diameter of the outer races. Smaller diameter rollers also distribute the load between a larger number of contact surfaces, decreasing the unit loads. Rollers and races are of chromium steel with 1 per cent carbon. Cages are one-piece bronze castings, having machined roller-aligning pads, and shaped to facilitate lubrication of the rollers. Spring-backed seals are used to exclude water and scale.

• New Adoptions of Materials and Processes

After intensive research, the Merkel Bobbin Company, have devised a combination paper and fabric base bakelite laminated bobbin head. In order to provide the necessary smoothness, the inside portion of the head is made of a paper base material, while to provide a high degree of impact resistance, the outer portion is a fabric base bakelite. Thus is attained a steaming bobbin with a head sufficiently strong to resist impact and a smooth surface that will not cause the yarn to tear.

All models of the new line of Westinghouse refrigerators have their cabinets Bonderized and then finished with Dulux. The reason given for using the Bonderizing

process is to preclude the possibility of the spread of rust should the finish be damaged accidentally, while Dulux was selected for the finish because it has an appearance like porcelain, is more impervious to high humidities, retains its lustrous appearance longer and offers more resistance to cuts and scratches.

Other features of the new line of refrigerators include a refrigerating top shelf in the froster, and on the inside of the door a tray that can be lifted from its supporting rack to convey drinks, desserts or other foods from the refrigerator to the dining room. A new design of door opener combines "finger action" with "knee action" so that a touch with either the finger or the knee will open the door.

Specifications for the 4-8-4 high-speed steam locomotives recently ordered by the Lehigh Valley Railroad from the Baldwin Locomotive Works include Timken roller bearings for all axles, including the driving axles. Timken bearings will also be used on the engine trucks of five other locomotives being built by Baldwins for the Lehigh Valley.

According to a recent report, Enduro stainless steel is now being used by Oliver United Filters, Inc., of Oakland, Calif., for such parts as studs, agitator hanger arms, small shafts and bushings in their filters. This material was adopted primarily for its corrosion resisting properties, these filters being used for various chemical and food industries, including sugar refining, and in gold mining where the filters are used in the cyanide process.

• Servo Principle Applied to Excavating Equipment

In a recently announced development of the Harnischfeger Corporation the "servo principle" is applied to the power clutch on excavators and cranes. It is claimed this "split-second" power clutch increases digging speed, prevents motor stalling and eliminates most of the physical labor of applying the clutch. It also reduces excessive strains and high maintenance cost on the excavators because of the speed with which the power can be released when the dipper strikes a heavy obstruction.

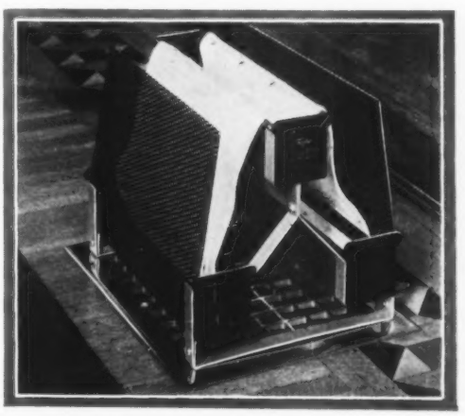
Referring to the accompanying picture of the power clutch, the large spur gear, the main clutch band, the clutch band engaging lever and the auxiliary clutch drum all rotate independently of the main drum. A pull on the operating lever causes rotation of the auxiliary drum to be retarded, thereby moving the clutch band engaging lever and forcing the clutch band against the drum.

Sufficient tension is maintained on the auxiliary clutch to hold the main clutch in its engaged position, while the release springs assure instant throwout of the auxiliary clutch and the main clutch when the operating lever is shifted to neutral position.

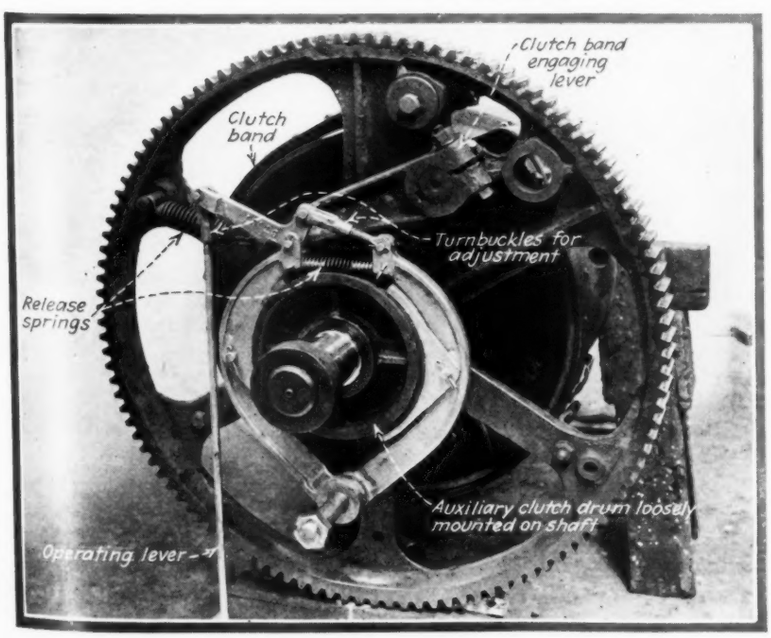
• Wicking for Evaporating Surfaces

In their development of a portable humidifier for placing on hot air registers to humidify residence rooms, engineers of the Taylor Instrument Companies encountered the problem of providing sufficient evaporating surface without resorting to pumps or spray nozzles. This was solved by using absorbent wicking, as shown in the accompanying picture of the finished product.

Three water pans with wicks arranged as shown give an evaporating surface of more than 200 sq. in. The ventilating plates are so arranged that the surfaces of the wicks are subjected to the maximum circulation of warm air, and the wicking material is of a special weave in order to give it the required absorbent properties. Under normal winter operation this device will add in excess of two gallons of water per day to the air in the room.



Portable room humidifier developed by Taylor Instrument Companies. All metal parts are brass and wicking is used to get maximum evaporating surfaces



Band type clutch using the "servo principle," as designed for cranes and excavators

... RUBBER ... For Impact and Abrasion

F. L. HAUSHALTER
Development Engineer, B. F. Goodrich Rubber Company

Important factors in designing rubber linings include size and shape of the impacting body, its depth of penetration, rubber thickness, the surface friction and the composition of the rubber

THE discovery that rubber is an excellent material to withstand abrasion is by no means a recent one. In 1736 La Condamine reported to the Paris Academy of Sciences that he had found natives on the banks of the Amazon wearing shoes made from a coagulated liquid which came from a tree called hevea. The discovery that rubber made a good protective sole was perhaps no more significant than the discovery in 1920 that a rubber lining in a pebble mill wore much longer than chilled iron in the wet grinding of ore.

Since these discoveries, much work has been done in the laboratory and in the field to find the best rubber composition to withstand abrasion and impact, as well as to find the proper design of the rubber parts to meet various service conditions. The resiliency of crude rubber and the toughness obtained by the addition of reinforcing pigments, such as carbon black, are the properties that make rubber highly resistant to abrasion and impact. Localized stresses set up by abrasive particles on a rigid material are greatly reduced on a rubber surface by flow or distortion.

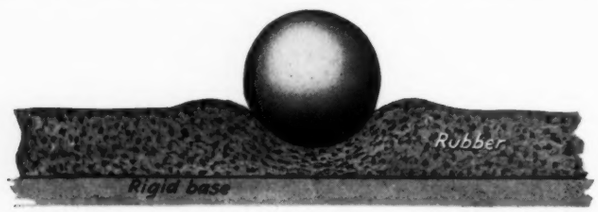


Fig. 1—When distorted under pressure, the thickness of the cushioning layer of rubber is reduced by displacement

In analyzing the abrasion-impact problem a falling ball is assumed to impound the rubber, as shown in Fig. 1. Since rubber, other than cellular material, is almost as incompressible as water, the volume of rubber displaced under a ball of a given diameter varies with the depth of penetration of the ball, as shown in Fig. 2. The volume displaced for specific amounts of penetration is not linear until after the depth of penetration equals the radius of the sphere.

Hardness Measurements

The system of comparative resistances to penetration as applied to metals is well known as the Brinell hardness number system. In this method a hardened steel ball is pressed into the material and the depth of the penetration is determined from the diameter of the depression by the formula $N = W/\pi Dh$, where W is load in kilograms, D the ball diameter in millimeters, and h the depth of penetration in millimeters. N is an expression of the unit pressure on the area covered by the ball.

Resistance to penetration is also a measure of the ability of rubber to withstand impact. In Fig. 3 the unit pressure on rubber, as determined from the Brinell hardness formula, is plotted against the penetration. In the curves, the penetration is given in terms of percentage of the original thickness of the rubber for various thicknesses of a grade of rubber used for chute linings. These curves show a rapid build-up of pressure for large deflections of the rubber. For the small size balls the rate of pressure increase changes little until the penetration into the rubber is 70 to 80 per cent of the thickness, whereas for the 4-in. ball the change in pressure increase is between 50 and 60 per cent of the rubber thickness.

The same data given in Fig. 3 is plotted in Fig. 4 against the load in pounds instead of pressure.

Distortion of the rubber sets up shearing forces in the horizontal plane. The magnitude of these forces depends to a large extent on how much rubber remains between the lowest point of ball contact and the solid base which supports the rubber.

Rubber is a vegetable and can be beaten into a pulp when the stresses set up by impact forces become ex-

cessive. For this reason the amount of deflection in the rubber is not as important as the cushioning thickness left under the ball. It is for this reason that the percentage deflection of the rubber rather than an arbitrary value of thickness determines the limitations for the various thicknesses.

Surface Friction

Another element which enters into the impact problem is that of surface friction. Part of the resistance to deflection is produced by the ball surface rubbing on the rubber. When rubber is dry the coefficient of friction is high, but when wet it is low. Rubber that stands up against impact when dry may fail in a short time if kept wet, because the deflection in the wet rubber is greater than in the dry rubber under impacts of the same intensity. This is particularly true when the impact force is near the limiting value for the particular thickness of rubber used.

On the other hand, abrasion of wet rubber takes place more slowly than dry rubber. This fact has established a place for rubber linings in various types of mills for wet grinding, whether the grinding is done by impact, attrition or slippage. Abrasion of rubber will also proceed slower in the absence of heat. An automobile tread will wear much longer on the same kind of road in winter than in summer.

In ball mill grinding, the particle size is reduced in three ways: by impact, attrition, and slippage. Ball mills operate most efficiently at certain critical speeds at which the crushing is done by impact. When operating speeds are decreased to less than the critical speed, grinding by attrition increases as the speeds are reduced. At speeds above the critical, part of the grinding is done by slippage. As determined by experiments at the Mines Experiment Station of the University of Minnesota, the critical speed N in terms of the radius R of the mill is given by the relation,

$$N = 54.19 / \sqrt{R}$$

The rate at which the grinding takes place depends on the diameter of the mill and the size of the balls, while the effectiveness of the cascading action is deter-

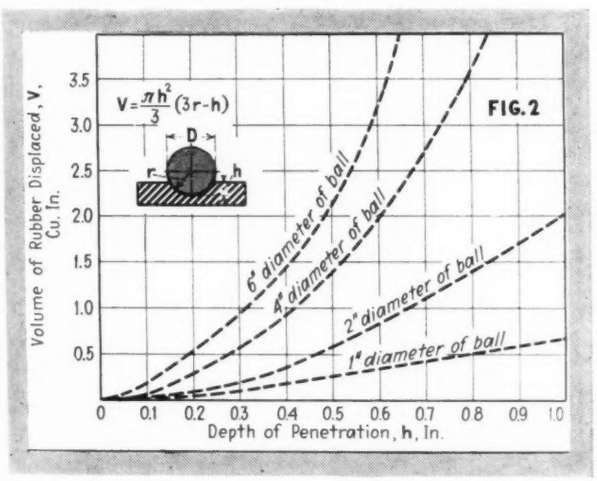
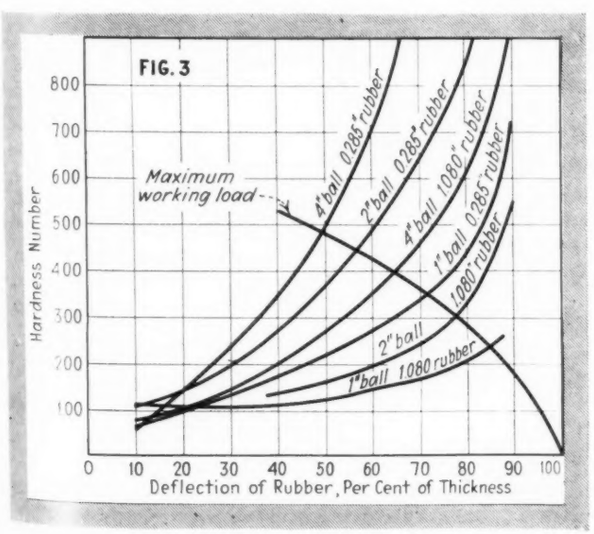


Fig. 2—Relation between displacement volume and depth of penetration for spheres 1, 2, 4 and 6 in. in diameter. **Fig. 3**—Effect of ball diameter and sheet thickness on the indentation hardness number of sheet rubber



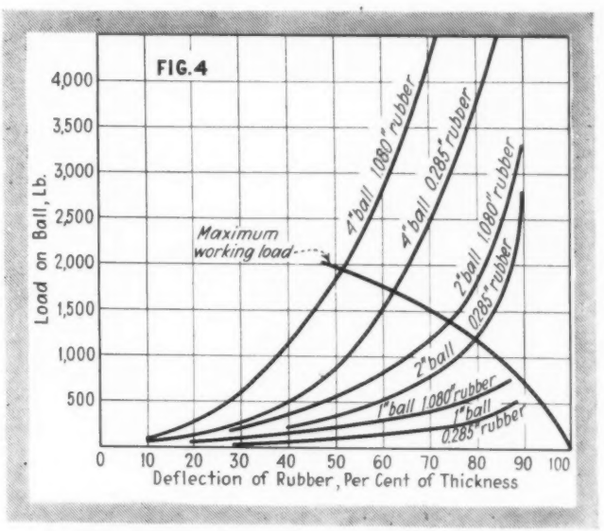


Fig. 4—Deflections in two thicknesses of rubber sheet for various loads using balls 1, 2 and 4 in. in diameter

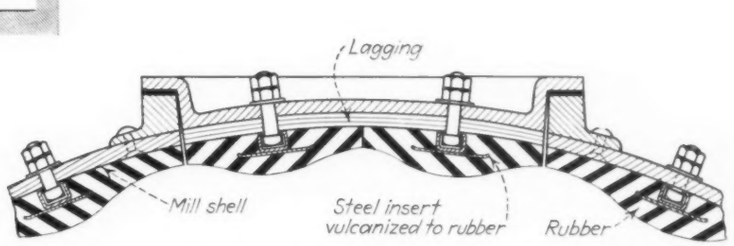


Fig. 5—Wave type lining used for abrasion resistance in rod mills for grinding sulphate paper pulp also protects the steel shell from corrosion

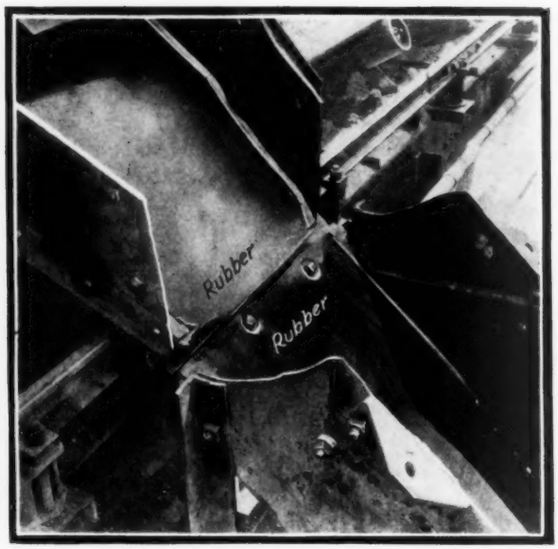


Fig. 7 — Rubber linings for chutes resist abrasion in handling ore, sand, gravel and limestone. Impact intensity, shape and hardness of the conveyed material are the factors determining the lining thickness

be applied and vulcanized without removing the mill from its foundation.

In rubber-lined ball mills there is usually a saving in replacement of grinding balls. Where cast-iron balls are used, impact with the steel linings breaks many balls, the broken pieces rapidly disintegrating. With rubber linings very few balls are broken.

Vulcanized rubber, securely attached to either a fabric-fiber backing or a steel backing, is used for lining chutes for ore, sand, gravel and limestone. The rubber is vulcanized to the backing at the rubber factory and the customer fastens the lining to the chute with rivets or bolts, as shown in Fig. 6.

Although the computation of the thickness of rubber

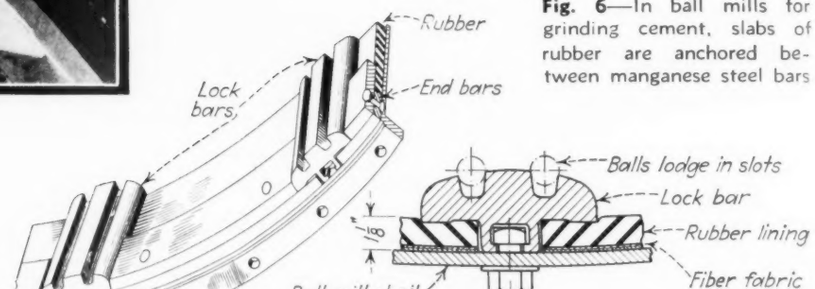


Fig. 6—In ball mills for grinding cement, slabs of rubber are anchored between manganese steel bars

mined by the speed of the mill and the height of the charge. Therefore, impact forces sustained by the rubber lining depend on this cascading action as well as the size of the balls and the speed of the mill.

Rubber is now used in impact-abrasion service in ball mills for the finish-end grinding of Portland cement, and also in mills for grinding ores, enamels, porcelains, and pulps of various kinds. Rod mills 6 ft. in diameter and 12 ft. long, for splitting the fibers of wood pulp have operated successfully for several years with rubber linings taking the blows produced by 50,000 lb. of 2½-in. diameter chrome steel rods rolling over each other at a mill speed of 14 r.p.m. As shown in Fig. 5, these rod mills use a wave-type rubber lining. The rubber slabs are 12 to 17 in. wide and are bolted to the shell of the mill by means of metal inserts which are securely bonded to the rubber. By keeping steel linings away from the sulphited pulp, rubber linings also prevent discoloration of the paper.

For the grinding of portland cement, the lock-bar type of rubber lining is used, as shown in Fig. 6. The slabs of rubber, having a fairly stiff fabric and fiber backing, are held to the mill shell by manganese steel retainer bars. To reduce wear in these bars, grooves are provided into which balls become firmly lodged, taking the impact and abrasion. Steel with 12 to 14 per cent manganese is used to resist abrasion and impact because this metal, somewhat like rubber, flows under impact. These bars run the length of the mill and are spaced about 14 to 18 in. apart.

To reduce contamination of the material being ground, the rubber can be vulcanized directly to the entire inner surface of the mill, as well as to the entire inner surface of the manhole cover. The rubber can

to be used to resist impacts of a certain order is based on the assumption that the falling bodies are spherical. spherical bodies are seldom found in practice. Further correction factors must therefore be introduced when sharp-pointed bodies are encountered. These factors depend to a large extent on the hardness of the material, for a sharp-pointed material that is soft will break up when it strikes the lining. A harder material produces a greater cutting action on the rubber. Correction factors governing recommendations for thickness of rubber where jagged material is to be handled must be determined to a large extent by experience.

New Lining Materials

Rubber compositions available for specific abrasion and impact applications are the result of years of research and experience. Eighteen years ago the best rubber compound had a tensile strength of about 2,500 lb. per sq.in.; today, rubber compounds are available with a tensile strength of 4,500 lb. per sq.in. This development has come chiefly through the use of reinforcing pigments and vulcanization agents in combination with improvements in processing machinery and methods.

Engineering data on the various types of rubber materials for specific uses are not of general value since a composition suitable for one application may not be desirable for a slightly different job. The insertion of one or two ingredients in the rubber may account for the success of the composition. As these recipes are usually held confidential by the rubber manufacturers, tabular information on the properties of rubber compounds would be of little assistance to the designer.

Charts for Obtaining Spring Dimensions Directly

Trial and error computations for obtaining desired maximum and minimum loads and lengths are eliminated in the procedure presented here

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ACCORDING to conventional methods the starting point in spring design is the maximum force on the spring from which the size, number of turns and the minimum force are computed. In the following method, not only the maximum force but also the minimum force can be used as a starting point in the design. This method has the advantage that all spring dimensions are determined directly without trial and error.

In any helical tension or compression spring working at a given maximum fiber stress, the ratio D/d, the mean diameter of the coil to the diameter of the wire, is fixed when the load ratios P_{min}/P_{max} and length ratios L_{min}/L_{max} are stipulated. Knowing the values of P or L, the dimensions D and d are immediately computable.

To facilitate the use of this method, charts shown in Figs. 1, 2, 3 and 4 have been prepared. The first two charts show the relation between length and force for compression and tension springs while the last two charts are used for finding safe maximum loads and deflections. The load chart, Fig. 3, has the ratios D/d added to assist in using this method. The deflection chart, Fig. 4, is based upon D/d values so that it becomes a single curve of deflection factors. In all these charts the increase in torsional stress caused by the curvature of the coils, as found by Wahl, is taken into consideration, and a maximum working stress of 50,000 lb. per sq.in. is used.

To illustrate the use of these charts, the compression spring shown in Fig. 6, is required to have a maximum force of 93 lb., a minimum force of 60 lb., maximum length of 6 in., and minimum length of $5\frac{3}{8}$ in. Assuming that the wire will be about $\frac{1}{4}$ in. in diameter, the net length from center to center of active coils will be $5\frac{3}{4}$ in. maximum and $5\frac{1}{8}$ in. minimum. The ratio of P_{min}/P_{max} is 60/93 or 0.645; and L_{min}/L_{max} is 5.125/5.75 or 0.892. On the compression spring chart, Fig. 1, these two values are followed to their point of intersection, this point being for a value of D/d equal to 5.9. The next

D = mean coil diameter.
d = wire diameter.
Length is measured center-to-center of wire between outer active coils. Minimum clearance between compressed coils is ***0.1d.***
Wahl factor is included in D/d ratios giving a maximum working stress of *50,000* lb. per sq.in. using *11,470,000* for torsional modulus.

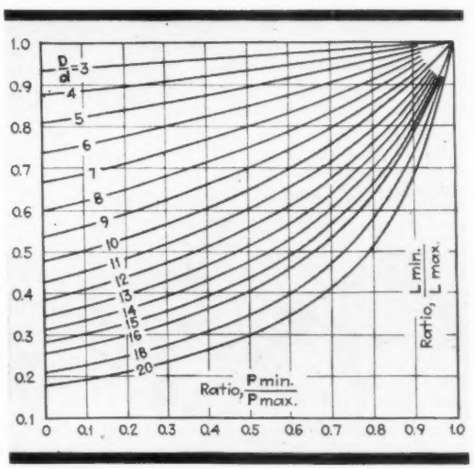


Fig. 1—Compression spring chart showing the ratio D/d for a given value of P_{min}/P_{max} and L_{min}/L_{max}.

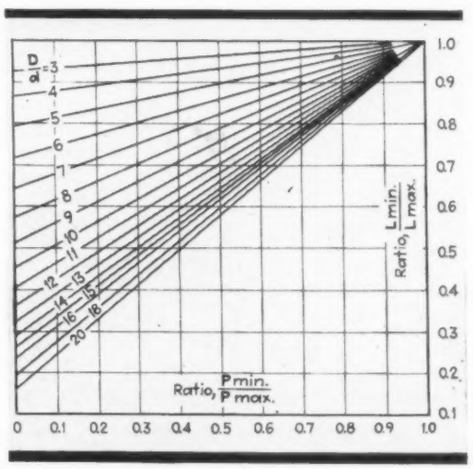


Fig. 2—Tension spring chart (to the right) giving the D/d ratio for any value of P_{min}/P_{max} and L_{min}/L_{max}

step is to find a spring having the ratio D/d equal to 5.9 and able to support 93-lb. maximum load.

Using the chart, Fig. 3, the point where the diagonal D/d line at value 5.9 intersects the diagonal load line at 93 lb., is projected to the left scale of the chart, indicating 0.190 in. wire diameter, and to the bottom of the chart indicating 1.12 in. mean coil diameter. The nearest standard wire size (listed at the right of the chart) is No. 6 W & M gage, or 0.192-in. diameter, which multiplied by the D/d ratio of 5.9 gives 1.13 in. for the mean coil diameter. This spring will have 60 lb. for the load at the maximum length.

The number of coils is found by dividing the minimum length of the spring ($5\frac{1}{8}$ in.) by the minimum pitch of the coils (0.211 in. wire diameter plus 10 per cent for clearance). This gives 24.29 or $24\frac{1}{4}$ active coils.

Free length or final dimension of the spring is found by multiplying the number of coils by the maximum deflection per coil, which yields the total deflection, and then adding this to the minimum length of the spring. The maximum deflection per coil may be taken from any spring table but it must be based upon the fiber stress of 50,000 lb. per sq.in. and should include the Wahl correction for curvature. Fig. 4 has been prepared to meet these requirements. At the value of D/d equal to 5.9 the maximum deflection per coil is 0.38 in. multiplied by d or 0.0729 in. per coil. Then $24\frac{1}{4}$ coils will deflect 1.77 in. This total deflection added to the minimum length ($5\frac{1}{8}$ in.) gives 6.895 in. as the free length of the spring from the center to center of active end coils.

Fig. 5 illustrates a tension spring designed to apply an approximately uniform torque to the shaft when the lever carried by it rotates between positions a and b. In this example the desired torque is 85 lb.-in. at both extremes with no decrease throughout the range of action. With the dimensions indicated, the forces required of the spring are 70.80 and 30.34 lb. at length 6.9 in. and 5.02 in., respectively.

The length ratio L_{min}/L_{max} is 0.727, and the load ratio P_{min}/P_{max} is 0.428. From chart for tension springs (Fig. 2) D/d is 8.9 and at this ratio, d in Fig.

Fig. 3—Maximum safe load, wire diameter, and mean coil diameter are obtained from this chart after the D/d ratio is found from one of the first two charts

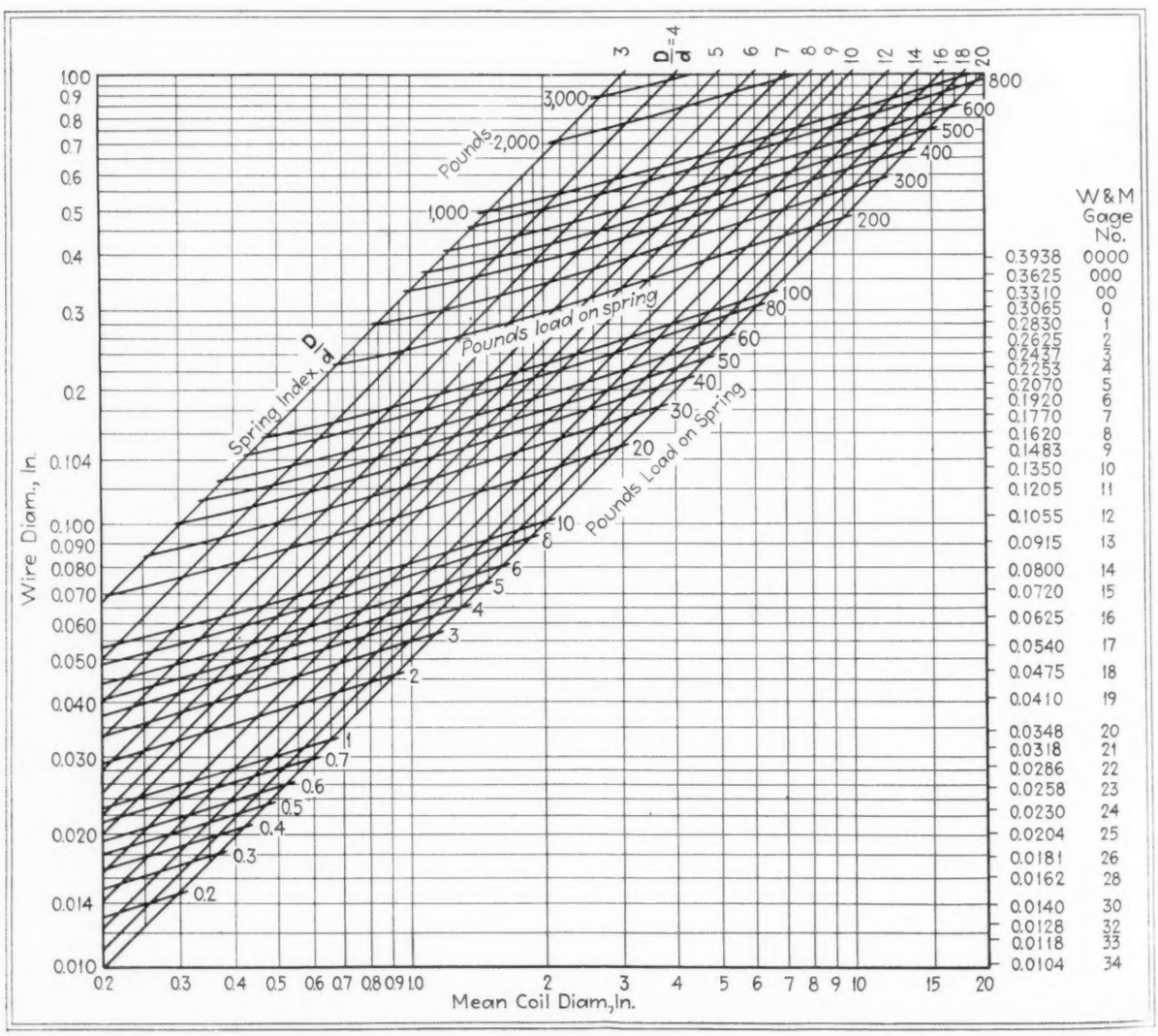
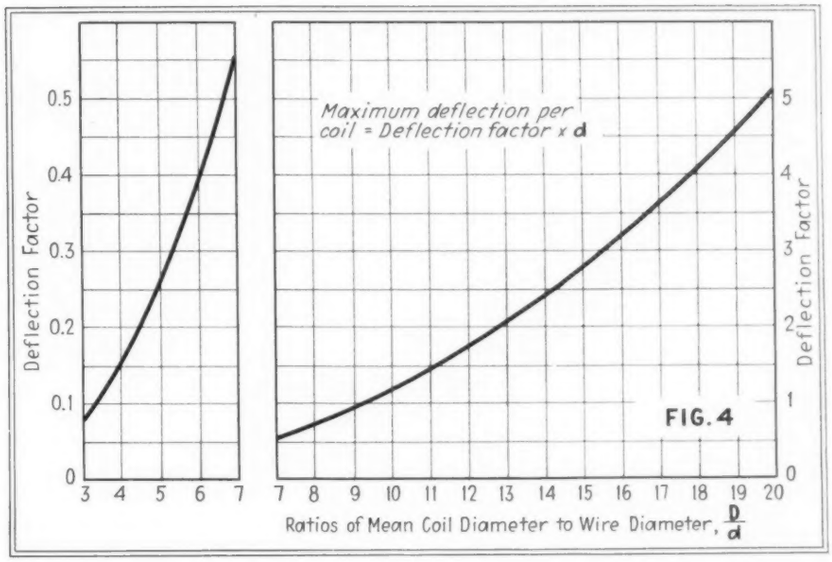


Fig. 4—Deflection per coil for a maximum fiber stress of 50,000 lb. per sq.in. is found from this chart for various values of D/d



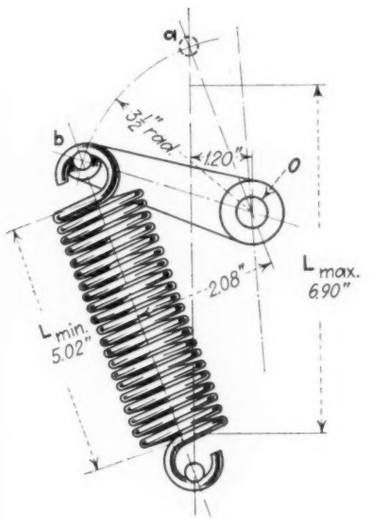


Fig. 5—Producing a uniform torque on crank arm, this tension spring can be designed entirely from the charts

3 is 0.207 in. or No. 5 W & M, and D is 1.83 in. diameter. The maximum deflection per coil is 0.195 in., maximum pitch is 0.402 in. and there are 17 coils.

Equations for the Charts

For the deflection per coil, Fig. 4, the maximum deflection f_{max} with round wire springs, in terms of the Wahl factor K and the maximum shear stress S, is given by:

$$f_{max} = S\pi D^2 / KGd \qquad (1)$$

Substituting C for the spring index D/d, and using values of 50,000 for S and 11,470,000 for G, Equation (1) divided by C^2 becomes:

$$f_{max} = 0.0137C^2d / K = K_ad \qquad (2)$$

Fig. 4 is the graph of Equation (2) for values of D/d from 3 to 20. Wahl factors used in preparing the chart are the ratios of actual shear stress to that computed by ordinary formulas.

Curves for compression springs in Fig. 1 are based on a clearance of $0.1d$ in the compressed position to avoid contact of coils. The maximum and minimum lengths per coil are:

$$L_{max} = 1.1d + f_{max} - f_{min} \qquad (3)$$

$$L_{min} = 1.1d \qquad (4)$$

Using Equation (2) for the value of f_{max}, the minimum deflection per coil in terms of the required spring forces is given by:

$$f_{min} = \frac{P_{min}}{P_{max}} \times f_{max} = K_ad\frac{P_{min}}{P_{max}} \qquad (5)$$

Substituting Equation (5) for f_{min} in Equation (3), the ratio of the minimum to maximum length becomes:

$$\frac{L_{min}}{L_{max}} = \frac{1.1}{1.1 + K_a - \left(\frac{P_{min}}{P_{max}} K_a\right)} \qquad (6)$$

Curves in Fig. 1 for compression springs represent Equation (6) plotted for several values of D/d. For tension springs the maximum length per coil from Equation (2) is:

$$L_{max} = d + f_{max} = K_ad + d \qquad (7)$$

From Equation (5) minimum length per coil is:

$$L_{min} = f_{min} + d = K_ad\frac{P_{min}}{P_{max}} + d \qquad (8)$$

The chart for tension springs, Fig. 2, is obtained by substituting various values of D/d in the following

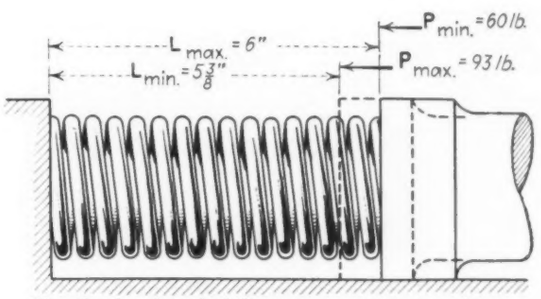


Fig. 6—Load and deflection requirements shown here for a typical compression spring can be determined without the use of cut-and-try methods

relation which is Equation (7) divided by Equation (8).

$$\frac{L_{min}}{L_{max}} = \frac{K_a\frac{P_{min}}{P_{max}} + 1}{K_a + 1} \qquad (9)$$

Factors depending on the number of coils do not appear in any of these equations, and the constant K_a contains terms applying to a single coil of either compression or tension springs.

Case Histories In Product Design–X

T. J. MALONEY
New Jersey Zinc Company

Brief descriptions of recent redesigns of three products that are used in the home, illustrating how careful attention to mechanical design and appearance increase salability

IN EVERY redesign of a product, the primary purpose is increased sales. There are only three engineering methods by which the market for any product can be increased, namely, reduced selling price, increased utility and an improved appearance. Frequently the last two of these factors go hand in hand. This applies to the three products described here.

1934 Kelvinator Refrigerators

Case History No. 27

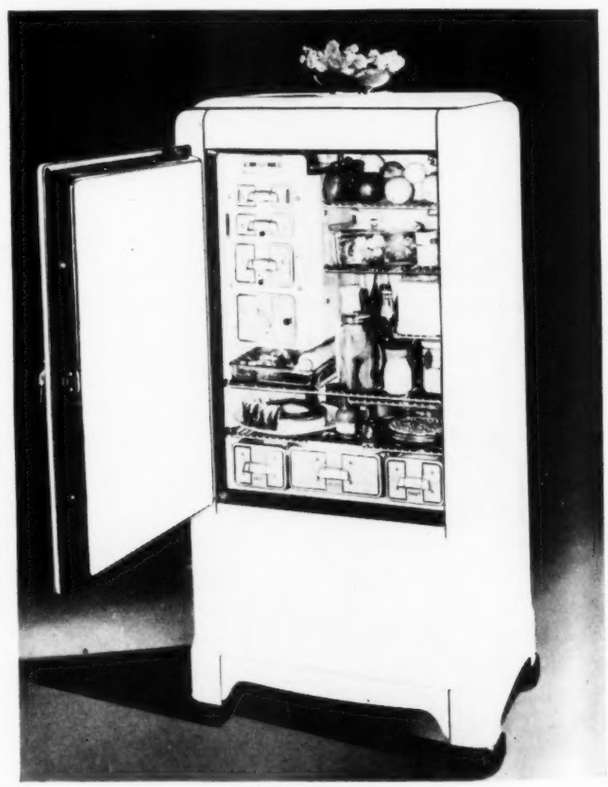
Created by the engineering department of the Kelvinator Company, the four lines of refrigerators described here represent the new 1934 domestic models that they have developed.

Design Aims—In common with all other machinery for the home, refrigerators must be both attractive and useful. But there is no piece of home equipment whose operation involves as many functions as those required of the electric refrigerator. This has called for the development of numerous automatic features, which had to be so designed that they would be simple and positive in their operation.

Thus, the problem of designing a modern domestic refrigerator is to develop a machine that will possess maximum convenience through automatic operation, be sturdy and dependable, and attractive in appearance.

Design and Manufacture—To meet every price class, four lines of refrigerators were developed. The "D" (DeLuxe) line contains six models, sizes ranging from 6.58 cu.ft., to 23.32 cu.ft.; the "P" line includes four models ranging from 5.13 cu.ft. to 8.73 cu.ft.; "S" line has three models, 5.29, 6.35 and 7.54 cu.ft.; and the "N" line also contains three models.

All of the models have cold-rolled, stretcher-leveled steel sheets for the exterior of the cabinet, the sheets being Bonderized to prevent rusting. The sides and back of the cabinet are made from a one-piece steel shell to give maximum rigidity, all of the corners being well-rounded. In all except the "N" models, the legs of the cabinets are integral with the side panels and have an inner core of sturdy spruce, thereby making the legs actually a continuation of the cabinet structural frame.



Porcelain finish is used for some of the new Kelvinators, lacquer finish for the others. Cabinet exterior is steel, Bonderized to prevent rusting

With this construction the absence of intervening joints eliminates the possibility of chipping the finish.

Bellows-type, quick make-and-break temperature controls are used. The DeLuxe models have four different zones of cold, automatically controlled; a frost chest for below-freezing storage; a super-fast freezing tray; fast-freezing trays and normal preservation temperature in the food compartment. In addition there is an automatic one-cycle return on special fast-freezing and defrosting and thermal overload protection for the motor.

Sales Success—Total shipments for the six-month period beginning October 1, 1933 were 75,885, as compared with 37,084 for the previous year and 42,383 for the six months between October 1, 1930 and March 31,

1931, which had been the best previous first-half period. The March, 1934 record of 30,009 units shipped compares with 16,143 units for the best previous March record, which was in 1933.

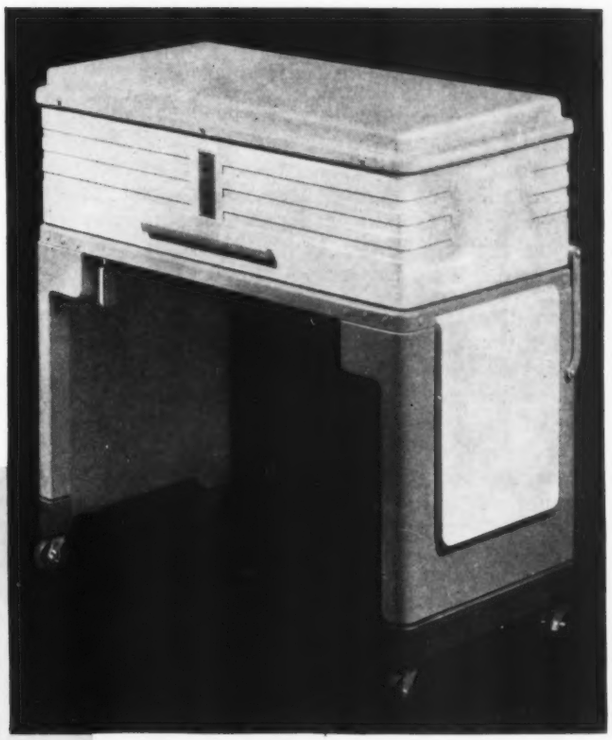
Probable Reasons for Success—It is never possible to determine the exact reasons for the success or failure of a merchandised product. But in every instance of success it is self-evident that the product must possess outstanding sales features. In this instance, not only was a tremendous effort put on the development of a new line of models, but the new models were backed up by a sales, advertising and sales promotion campaign which has been the biggest in the history of the company.

The Easy Ironer

Case History No. 28

Mechanical design of the new ironer manufactured by the Easy Washing Machine Corporation was developed under the direction of their chief engineer, P. Edward Geldhof, with Van Doren and Rideout as consultants on the appearance of the product.

Design Aims—Both appearance and utility were considered of major importance in this redesign. It was



Transformation of the Easy ironer from a household machine to a piece of home equipment furniture by "packaging" it in an attractive and serviceable housing

decided that in addition to being highly efficient and easy to operate, the ironer should be designed so that when not in use, it was to be completely inclosed, giving the appearance of a piece of kitchen furniture and serve as a table. Thus the redesign was primarily a problem of "packaging."

Design and Manufacture—The accompanying pictures of the new model and its predecessor clearly show the results. The top of the ironer is given a setback appearance. In addition to contributing to the appearance, the horizontal stripes embossed in the cover stiffen and strengthen that part.

One particularly interesting feature is the manner in which the name is applied, the old stenciled effect having been discarded.

The principal functional improvement in this product is the type of heating element used and the placing of a shelf at each end of the ironer. One shelf is for the clothes that are ready to be ironed and the other is for those already ironed.

Green, a favorite color for the kitchen, was selected for the main color of this machine. A three-tone effect was used, giving both an appearance of cleanliness and a soft colorful touch. Other colors are obtainable as special.

Because of the nature and extent of the changes, the cost of the new machine is slightly greater than that of the previous model. But this higher cost was deemed to be more than compensated for by the greater sales value of the new model.

With reference to the increased cost, it should be stated that no excessive tool cost was involved and no

new manufacturing equipment was required, the increased cost being reflected directly in the product in the form of additional material and labor.

Sales Success—Because this machine has been on the market for a relatively short time, it is too early to predict its commercial success. However, present orders are well ahead of estimated production and salesmen and retailers are enthusiastic about it. At its initial showing at the Chicago House Furnishings Show it was accorded a cordial reception.

Probable Reasons for Success—Probably the basic reason for the apparent success of this product is the thoroughness with which the problem was studied previous to starting the design. This included a thorough study of the trend in kitchen equipment, a review of the new ideas in housing construction, and a recognition of the fact that in modern homes, space is at a premium. A second factor in the development of this product was that all designs and details were developed with a full recognition of the production tools and equipment available, thereby virtually eliminating new tooling and equipment charges. Finally, the close cooperation between the engineering department and the industrial designer was also a major factor.

Friez Humidistat and Thermostat

Case History No. 29

The development of the two instruments described here was carried out entirely by the engineering staff of the Julian P. Friez & Sons, Inc.

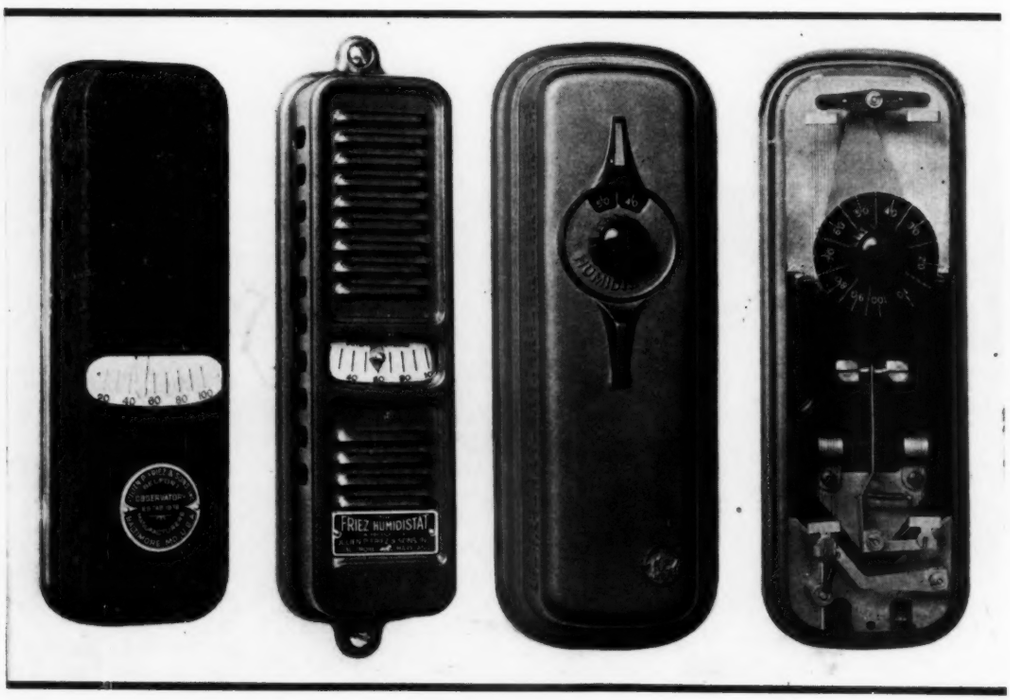
Design Aims—With a large potential market for humidistats and thermostats being created by the rapid entry of air conditioning into the home, public buildings, and railroads, it was apparent that the appearance of the old instrument designed for industrial application would not be suitable. For this reason, it was decided to redesign these instruments, making not only improvements in the mechanism, but also giving the instruments a pleasing appearance and finish.

Design and Manufacture—Improvements in the design and finish of the cases in which the humidistats and thermostats are housed are evident in the accompanying illustrations which show the latest models and two preceding ones.

The oldest of the instruments shown here was produced up until a little more than a year ago. The case was finished in black crystalline lacquer with the internal parts made from brass and machined bakelite. In the second model shown here can be seen an improvement in the form of the case, much better ventilation having been provided. In this second design, the case is made of sheet steel, cadmium plated to prevent rusting and increase the durability of the color lacquer finish. In this second design, the internal parts were made of brass stampings acid dipped and lacquered.

In the latest model humidistats, illustrated here, the cases are drawn out of pure aluminium and given the Alumilite or anodic treatment which not only renders them beautiful, but gives a surface finish that is proof against corrosion from all usual atmospheric conditions, moisture and scratches. The nameplate has taken the form of a coined medallion located off center near the bottom. The mechanism of the instrument is mounted on a sub-base of molded bakelite with metal insert plates. All the smaller metal parts are cadmium plated to prevent corrosion.

Sales Success—Deliveries on the new items have just begun, and as reported by the manufacturer "A tremendous amount of business and inquiries have been received already."



Three steps in the development of the latest model of Friez humidistat and thermostat for heating and air conditioning. On the extreme right is shown a humidistat with the cover removed. The latest model has an Alumilited aluminum case with the skirt carried down to cover the molded bakelite base on which the mechanism is mounted

Determining the Critical Speeds of Shafting

Rotational speed at the natural frequency of vibration for a shaft under various loadings and types of support can be determined directly by one of the following eight solutions

WILLIAM C. WILLARD
Chief Draftsman, Buffalo Forge Company

IN DETERMINING the critical speed of a horizontal shaft the major operation is to determine the maximum deflection of the shaft caused by its own weight and the weight of pulleys, couplings, fans, motor rotors or other loads mounted on the shaft. Although the shaft deflection is not a direct function of the critical speed, once the deflection is known, the first critical speed n_c in r.p.m. can be determined by the following formula:

$$n_c = 187.7/\sqrt{y} \qquad (1)$$

Of the eight cases to follow, the simplest one is that of a straight shaft of uniform diameter d carrying one concentrated load placed midway between the bearings. As shown in the diagram, Fig. 1, and the equations included, the total maximum deflection y is the sum of y_1, the deflection caused by the concentrated load W, and y_2, the deflection caused by the weight of the shaft. The effective load of the combined shaft weight and concentrated load produces the maximum deflection given by the following relation:

$$y = 0.000,001,46\left(\frac{l^3}{d^4}\right)\left(W + \frac{5S}{8}\right) \qquad (2)$$

This value substituted in Equation (1) gives the first critical speed.

When the concentrated load is not located midway between the bearings, but at a distance a from the left bearing, as shown in Fig. 2, the deflection caused by the load alone is given by the equation for y_1 in Fig. 2. The deflection at a distance a caused by the combined concentrated load and shaft weight is:

$$y = 0.000,002,92\ \frac{ab}{ld^4}\left[8Wab + S(l^2 + ab)\right] \qquad (3)$$

When the load is not far off center, the deflection at the center caused by the weight of the shaft is only slightly greater than the deflection at the load. For such designs the value of y_2, given in Fig. 1, can be combined with the value of y_1, given in Fig. 2, giving an approximate value of the deflection with sufficient accuracy for most purposes.

For a straight shaft with one concentrated load overhung beyond one bearing, as shown in Fig. 3, the deflection y_1 caused by the load, and the deflection y_2 caused by the shaft weight are both given by equations accompanying this diagram. Combining these two de-

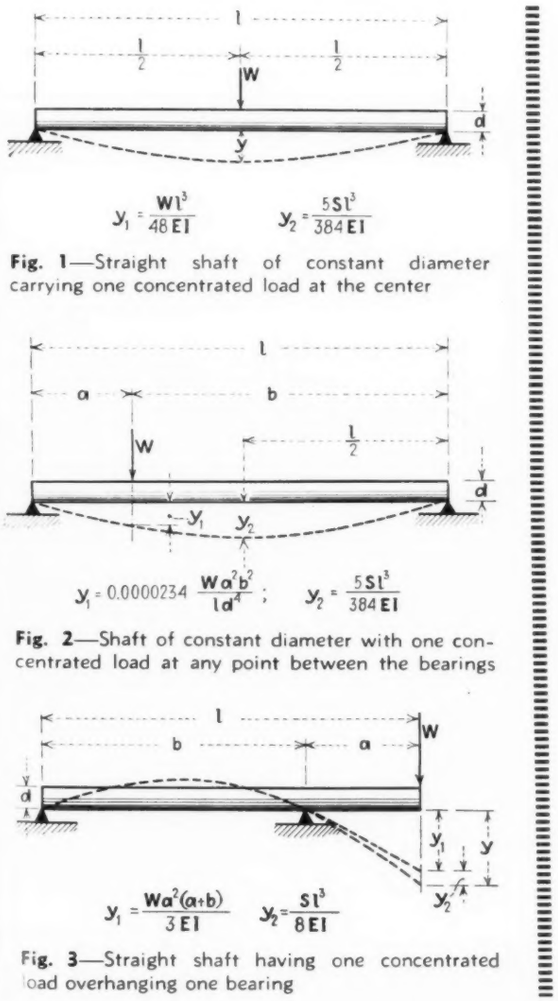


Fig. 1—Straight shaft of constant diameter carrying one concentrated load at the center

Fig. 2—Shaft of constant diameter with one concentrated load at any point between the bearings

Fig. 3—Straight shaft having one concentrated load overhanging one bearing

flections, for the case when the overhand a is small compared to the distance b between bearings, the total deflection in terms of the elastic modulus E and moment of inertia I becomes:

$$y = \frac{l}{EI}\left(\frac{Wa^2}{3} + \frac{Sl^2}{8}\right) \tag{4}$$

More accurate results are obtained by the use of the following equation for the total deflection when a is not small compared with b:

$$y = 0.000,001.46\ \frac{l^3}{d^4}\left(\frac{l}{b}\right)^2\left[W + \frac{5S}{8}\right] \tag{5}$$

A shaft with two diameters carrying one overhung load and one midway between the bearings, as shown in Fig. 4, can be calculated in the following manner: The effective loads, L_1, and L_2 at W_1 and W_2, where S_1 and S_2 are the weights of the shaft having diameters d_1 and d_2, are given by:

$$L_1 = W_1 + \frac{5S_1}{8}$$

$$L_2 = W_2 + \frac{5S_2}{8}$$

Assuming M_1 and M_2 to be the masses corresponding to L_1 and L_2, the relation of the two masses is the same as the ratio of their weights. Substituting δ for the ratio M_2/M_1 and for the ratio I_2/I_1, the first critical speed n_{c1} and the second critical speed n_{c2} are given by the following equations, the values of the constance β_1 and β_2 being determined as shown in Fig. 4:

$$n_{c1} = \frac{30\,\beta_1}{\pi}; \quad n_{c2} = \frac{30\,\beta_2}{\pi} \tag{5}$$

This case is one commonly encountered in determining the citical speed of shafts for electric motors.

When a shaft of constant diameter carries three or more loads between the bearings as shown in Fig. 5, the Dunkerley formula can be simplified to the following form. In this equation y_1, y_2, y_3, etc., are the deflections caused by W_1, W_2, W_3, etc., as determined by Equations (2) and (3) given above.

$$n_c = \frac{30}{\pi}\sqrt{\frac{386}{y_1 + y_2 + y_3 + \ldots}} = \frac{187.7}{\sqrt{\Sigma y}} \tag{6}$$

This equation is for concentrated loads only. Therefore, an additional deflection factor must be included to take into account the effect of the shaft weight upon the shaft deflection. The deflection of a load at the center of the shaft equivalent to the weight of the shaft is determined from equations included in Fig. 1 and Fig. 2, this deflection being added in Equation (6).

A shaft having two or more diameters, but carrying one load at the center and having the same reaction at both bearing supports, is shown in Fig. 6. By assuming the shaft to be clamped rigidly at the point where the central load is located, the reaction at the left support A, treated as a concentrated load on a cantilever beam, is given by:

$$R = 50\left(W + \frac{5S}{8}\right)$$

The deflection under this load is given by the following equation:

$$y = \frac{R}{4.263}\left[\frac{l_1^3}{d_1^4} + \frac{l_2^3 - l_1^3}{d_2^4}\right] \tag{7}$$

For the general case of a shaft having any number of diameters with one load at the center, as shown in Fig. 7, the deflection formula becomes:

$$y = \frac{R}{4.263}\left[\frac{l_1^3}{d_1^4} + \frac{l_2^3 - l_1^3}{d_2^4} + \frac{l_3^3 - l_2^3}{d_3^4} + \ldots \frac{l_n^3 - l_{n-1}^3}{d_n^4}\right] \tag{8}$$

For a shaft having any number of concentrated loads between the bearings with the shaft having any number of diameters, as shown in Fig. 8, the reactions at each suport are found for each load. Each reaction R is based on the combined load of W and the weight of the shaft between W and R. The deflection y_1 for the section to the left of the load W_1 is obtained by the use of Equation (8). Deflection y'_1 for the section to the right of the load W_1 is found by substituting the corresponding values for l, d and R. For each of the loads:

$$y_1 = \frac{R}{4.263}\left[\frac{l_1^3}{d_1^4} + \frac{l_2^3 - l_1^3}{d_2^4} + \frac{l_3^3 - l_2^3}{d_3^4}\right] \tag{9}$$

$$y_2 = \frac{R}{4.263}\left[\frac{l_1^3}{d_1^4} + \frac{l_2^3 - l_1^3}{d_2^4} + \frac{l_4^3 - l_2^3}{d_3^4}\right] \tag{10}$$

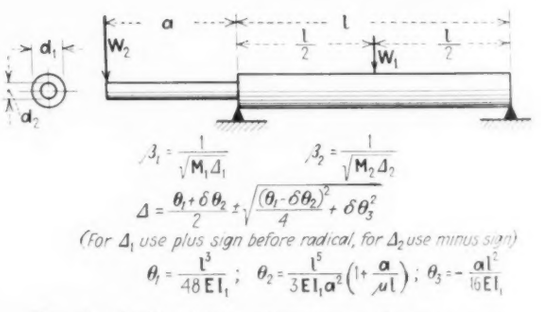


Fig. 4—Shaft of two diameters with one overhung load and one load midway between the bearings

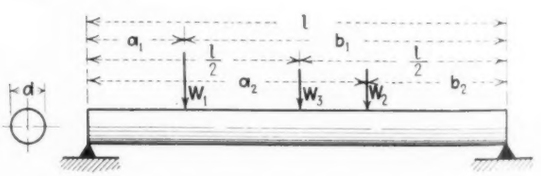


Fig. 5—Three or more loads between bearings on a shaft of constant diameter

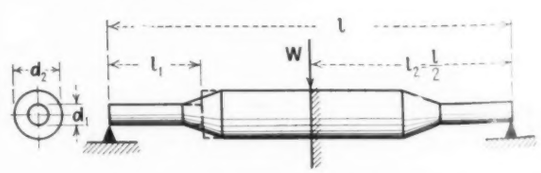


Fig. 6—Shaft of two or more diameters carrying one load at the center

$$y_3 = \frac{R}{4.263}\left[\frac{l_1^3}{d_1^4} + \frac{l_2^3 - l_1^3}{d_2^4} + \frac{l_3^3 - l_2^3}{d_3^4}\right] \quad (11)$$

$$y_1' = \frac{R}{4.263}\left[\frac{l_{10}^3}{d_5^4} + \frac{l_9^3 - l_{10}^3}{d_4^4} + \frac{l_6^3 - l_9^3}{d_3^4}\right] \quad (12)$$

$$y_2' = \frac{R}{4.263}\left[\frac{l_{10}^3}{d_5^4} + \frac{l_9^3 - l_{10}^3}{d_4^4} + \frac{l_7^3 - l_9^3}{d_3^4}\right] \quad (13)$$

$$y_3' = \frac{R}{4.263}\left[\frac{l_{10}^3}{d_5^4} + \frac{l_9^3 - l_{10}^3}{d_4^4} + \frac{l_8^3 - l_9^3}{d_3^4}\right] \quad (14)$$

To find the true maximum deflection, the deflection to the left of W_1, Equation (9), is paired with the deflection to the right of W_3, Equation (14). The true deflection for the pair is determined from the construction shown in Fig. 9. In this diagram the two calculated deflections y_1 and y'_3 are drawn to scale at the reactions, y_A corresponding to y_1, and y_B corresponding to y'_3. The true deflection z at the load is,

$$z = y_B + (y_A - y_B)\frac{b}{l} \quad (15)$$

True deflections are found for each of the three loads. For W_2, the deflections are given by Equations (10) and (13), and for W_3, Equations (11) and (12) are used. The maximum deflection (the sum of the true deflections) when substituted in Equation (1) gives the first critical speed.

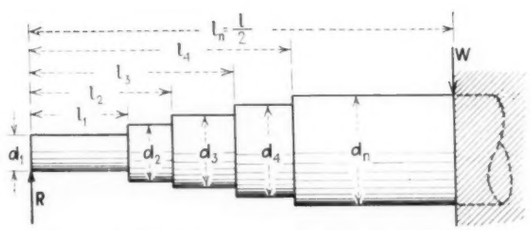


Fig. 7—General case for a shaft of any number of diameters supporting one load at the center

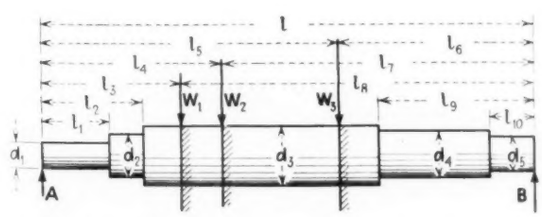


Fig. 8—Any number of concentrated loads supported by a shaft having any number of diameters

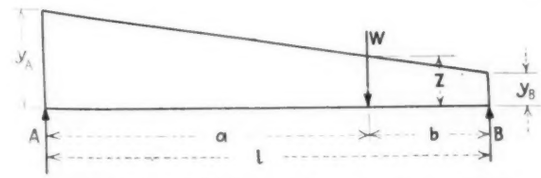


Fig. 9—True deflection z for any pair of right and left-hand deflections is given by Equation (9)

Load-Carrying Ability of Metal Spur Gears

ACCORDING to the Recommended Practice of the American Gear Manufacturers' Association, in the design of spur gears, the beam strength of the gear tooth must be sufficient to withstand the maximum momentary load set up by the operating or dynamic conditions, and it is calculated by the equation:

$$W_d = \frac{0.05 \times PLV \times (FW \times C + W)}{0.05 \times PLV + \sqrt{FW \times C + W}} + W$$

W_d = total dynamic load, lb.
FW = face width of gears, inches
W = 33,000 × hp./PLV = static tooth load
PLV = ft. per min. pitch line velocity
C = deformation constant from Table I

To use the foregoing equations, it is necessary to have some measure of the error in action of the gears. Classifications and tolerances for the errors in gears have therefore been set up by the A.G.M.A. For this purpose, gears are grouped in three classes, Class I for well-cut commercial gears; Class II for accurately cut commercial gears, and Class III for accurately ground gears. In Table II are given the maximum errors corresponding to various diametral pitches.

Noise of operation is usually a good test of the accuracy of gears, and the permissible inaccuracy of the gears is usually based on the noise they make. The allowable errors given in Table III should keep the noise of operation and the intensity of the dynamic load within reasonable limits. Where extreme quietness is desired, a higher order of accuracy is required.

Table I—Deformation Constant *C*

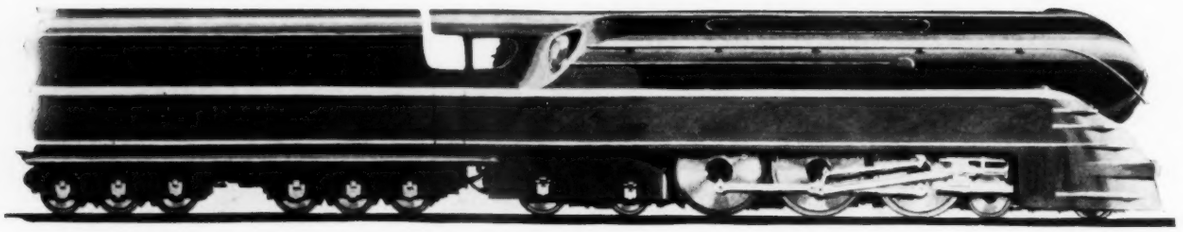
Materials	Tooth Form	Value of C			
		.0005	0.001	0.003	0.005
C. I. & C. I.	14½ deg.	400	800	2400	4000
C. I. & Steel		550	1100	3300	5500
Steel & Steel		800	1600	4800	8000
C. I. & C. I.	20 deg. full depth	415	830	2490	4150
C. I & Steel		570	1140	3420	5700
Steel & Steel		830	1660	4980	8300
C. I. & C. I.	20 deg. Stub	430	860	2580	4300
C. I. & Steel		590	1180	3540	5900
Steel & Steel		860	1720	5160	8600

Table II—Maximum Error in Spur Gears

DP	Class I	Class II	Class III
1	0.0048	0.0024	0.0012
2	0.0040	0.0020	0.0010
3	0.0032	0.0016	0.0008
4	0.0026	0.0013	0.0007
5	0.0022	0.0011	0.0006
6 or more	0.0020	0.0010	0.0005

Table III—Allowable Error in Spur Gears

PLV	Error	PLV	Error	PLV	Error
250	0.0037	1750	0.0017	3250	0.0009
500	0.0032	2000	0.0015	3500	0.0008
750	0.0028	2250	0.0014	4000	0.0007
1000	0.0024	2500	0.0012	4500	0.0006
1250	0.0021	2750	0.0011	5000	
1500	0.0019	3000	0.0010	and over	0.0005



Streamlining the Railroads

A brief historical survey of streamlining and the fundamental changes that it is now making in the engineering principles being applied in the design of vehicles

O. KUHLER
Design Engineer

THE vogue of streamlining we are experiencing today has its roots in the past. Even when aerodynamic science was practically unknown, streamlining was used extensively on all kinds of transportation vehicles, but it was not called "streamlining" then. When speaking of the fleeting lines in the hull of a clipper ship or the reposing curves in a stage coach, it was called "graceful lines." Today the layman, unfamiliar with aerodynamics, calls almost everything "streamlined" when he really means "graceful lines." We have "streamlined" radio cabinets, toasters, cigarette lighters, and even gasoline with "streamlined" action.

Streamlining As a Decorative Element

The development of the science of aerodynamics and its application to airships and airplanes has created in the public a sense for fleeting lines and these lines, being recognized by designers as a decorative element, have been pronounced to give the impression of speed that correctly aerodynamically streamlined bodies possess.

The automobile manufacturer, to give the sales appeal of a visual impression of the speed that the engineer has built into the machine, has used and is using "streamlining" extensively. Not always has this been done effectively and freaks that the public rejected have appeared. Such freaks have usually been the result of transferring to a wheeled vehicle bound to the ground the lines and principles scientifically developed for bodies moving in free air. Such a procedure is wrong in principle and therefore does not give the desired beauty.

An entirely different kind of aerodynamic and decorative streamlining must be developed for vehicles moving on the ground. Varying speeds, varying power output and varying adhesive friction constantly alter the requirements. While in a free-moving, free-flying body we may go the limit for aerodynamic efficiency, the above-mentioned variable factors in ground vehicles demand a more conservative development, largely governed by practical demands of serviceability.

Serious efforts taking those facts into consideration are being made now in the streamlining of railroad cars, automobiles and locomotives, and the results will show a limited aerodynamic efficiency compared to free-flying objects. They will, however, show a relatively great aerodynamic improvement compared to conventional vehicles and an improved appearance with reference to logical beauty.

There is nothing new about streamlining railroad equipment. In the late nineties a streamlined train, known as the "Adams Wind-splitter" was tried out on the Baltimore & Ohio Railroad. The records indicate that the train made one run between Baltimore and Washington, covering the distance of 36.8 miles in 37 minutes. Curiously enough "windsplitting" shapes were applied only to the coaches, while the engine was of standard form.

New Design Principles Being Applied

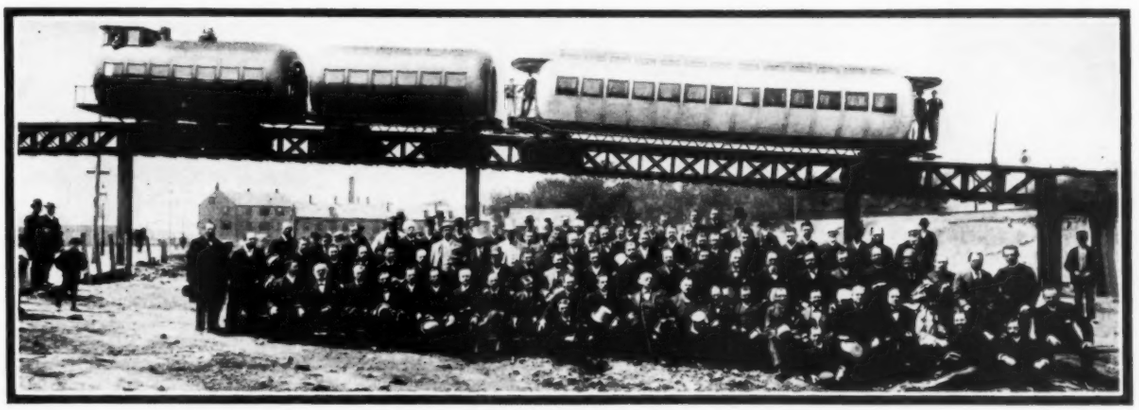
An entirely inclosed engine of good appearance, with the engineer in front and the tender covered like the engine, of German design, was shown in 1876 at the World's Fair in St. Louis. This engine made a few fast runs, but was probably ahead of its time. In some of the designs of locomotives of today the practice to place engineer at the front has been tentatively revived. But this never finds much encouragement from the engine driver on account of almost certain death to him in case of a serious collision.

While the standard engine along the lines of the

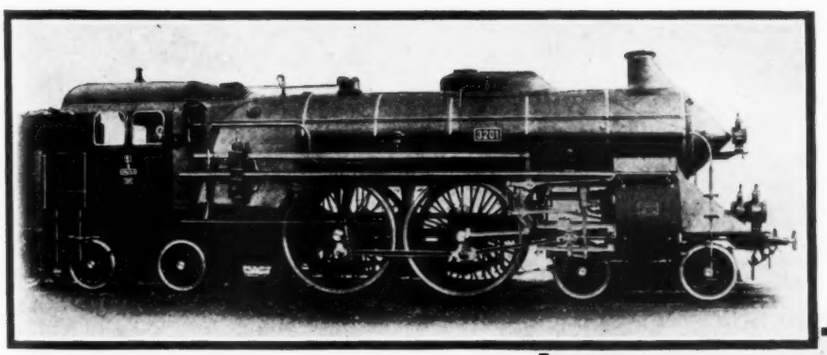
Adams "Wind-splitter" looks out of place, the same will be true if a thoroughly streamlined motive-power unit is attached to a train of modern standard equipment. Light-weight coach construction points toward a tubular form or monocoque construction for greatest structural strength and this is bound to influence outside appearance. To reduce skin friction the windows will have to be set flush with the outside walls, which in turn will make for easier maintenance. Aerodynamically, a real problem exists to close up the space between the vestibules of cars. While this problem is not so difficult in articulated trains, where a fixed flexible joint may be used, it will be harder in trains that have to be lengthened or shortened according to capacity desired. Engineers will solve it with modern fabrics. Aerodynamically, this is one of the weakest points in the streamlining of a train.

In the airplane the industrial designer has learned how to use structural members also as decorative elements. This new possibility is responsible for having taught us to see beauty and gracefulness in a purely practical and mathematically devised form. It is this new sense of appreciation which will largely govern the decorative scheme of modern car interiors. Structural parts, previously covered and hidden behind a wood panel with ornaments on it, will be exposed and interwoven into the decorative scheme. This will save weight, which is now recognized as of importance in the operation of railroads.

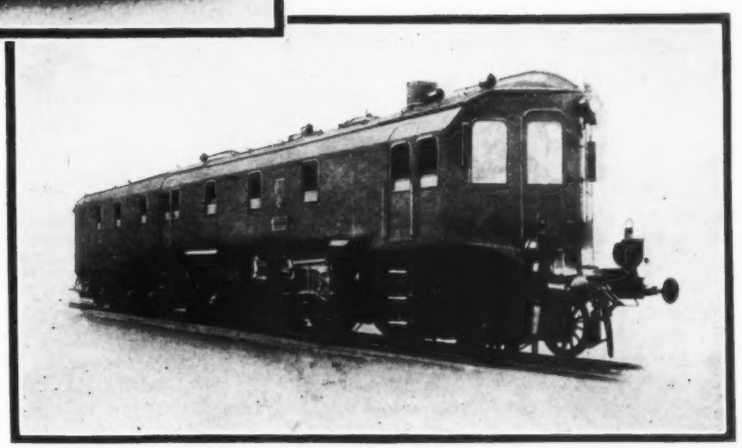
Correct streamlining of the tail end or observation car gives a splendid opportunity for improvement. The "rear porch" as in use today does not satisfy us artistically or aerodynamically. It will be entirely inclosed and have a different seating arrangement, allowing many more travelers to sit at the one spot of the train from which, with correct design of windows, an overall view of the landscape can be obtained. This is a valuable feature in bidding for passenger traffic.



Reception to the Philadelphia City Council by the Meigs Elevated Railroad Construction Company at East Cambridge, Mass., on May 27, 1887. This steam powered train illustrates the first application of streamlined and tubular construction. The trucks and track construction are unique



Built in 1904 by Henschel & Son of Cassel, Germany, this 4-4-4 locomotive represents the first attempt at streamlining that type of equipment. This locomotive was operated on the Prussian State Railroads, and in a trial run reached a speed of more than 80 m.p.h., a record of the time



Four-cylinder locomotive built in 1912 by J. A. Maffei of Munich, Germany, and operated on the Bavarian State Railroads. With its 86 in. diameter driving wheels, this locomotive reached a speed of almost 93 m.p.h., an all-time record

COMMUNICATION and COMMENT

Form Generator for Gear Teeth

A. C. RASMUSSEN
Insley Manufacturing Company

● Form factors for the design of gear teeth using the original Lewis formulas are not available for long-addendum pinion teeth or short-addendum gear teeth. For such designs, the form factor is ordinarily determined from a large scale layout of the gear, from four to ten times actual size, in order to obtain sufficient accuracy.

To avoid making these large-scale layouts, the tooth-form generator, shown in Fig. 1, was developed by the author for use on the drawing board. By means of this instrument a large-scale tooth form can be developed for a gear with any number of teeth, without drawing in the pitch circle or gear center.

As shown in Fig. 1, the device has an "axis" hole near the right edge. Centered at this hole, the curved edge at the left carries lines marked off in degrees above and below a horizontal line passing through the axis. In a rectangular hole in the center of the instrument, a tooth-form templet is secured, adjustable for the desired addendum.

In using the instrument a layout is made over horizontal and vertical center lines, as shown in Fig. 2, the intersection being the point where the tooth center line crosses the pitch circle.

To determine the scale of the pitch-diameter, the number of teeth in the gear is divided by 0.30, the diametral pitch of the instrument. The scale length of the pitch radius is then multiplied by the x and y values given in the table for 5-deg. increments of rotation. These y factors are plotted downward from the axis, and x factors marked off horizontally to either side of the y points.

Placing the instrument over the intersection of the center lines (the zero rotation point) the instrument is set so that zero deg. on the protractor scale lies over the horizontal line drawn through the zero rotation point. Along the edge of the tooth form templet a line is drawn on the paper.

The instrument axis is placed over each of the other plotted rotation points on the left, setting the lower protractor scale in each instance with the rotation angle matching the horizontal line through the plotted point. In Fig. 2, the instrument is in the position corresponding to 20-deg. rotation. Lines are drawn along the tooth-form templet for each of these settings, both to the right and left. For the points on the right, the upper protractor scale is used.

With these series of tangent lines as a guide, the tooth form can then be drawn in with an involute curve, and

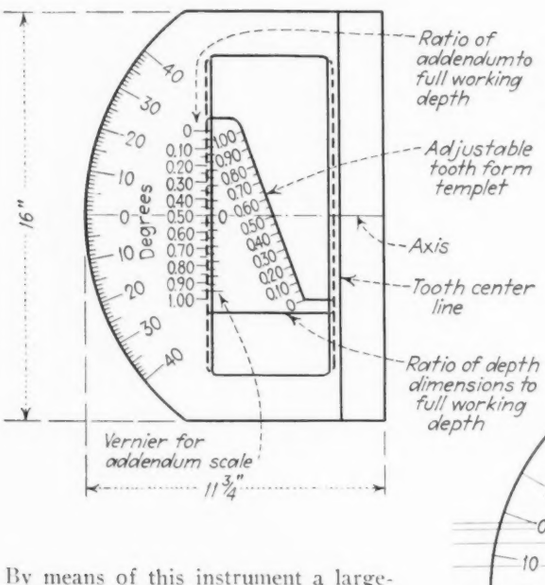


Fig. 1—Any tooth form can be drawn with this instrument to a large scale without making a layout of the gear

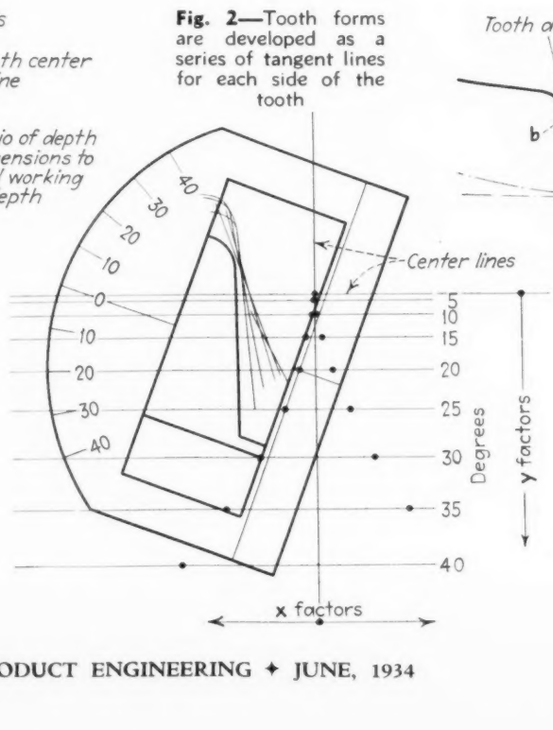


Fig. 2—Tooth forms are developed as a series of tangent lines for each side of the tooth

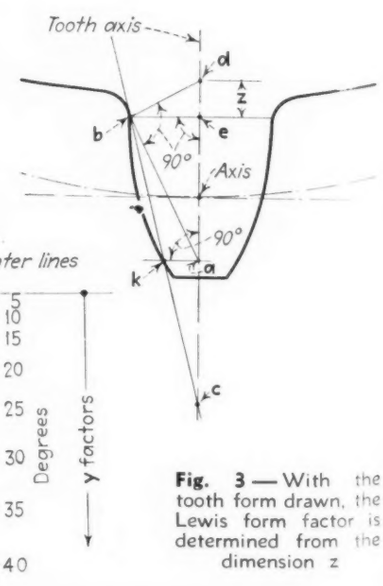


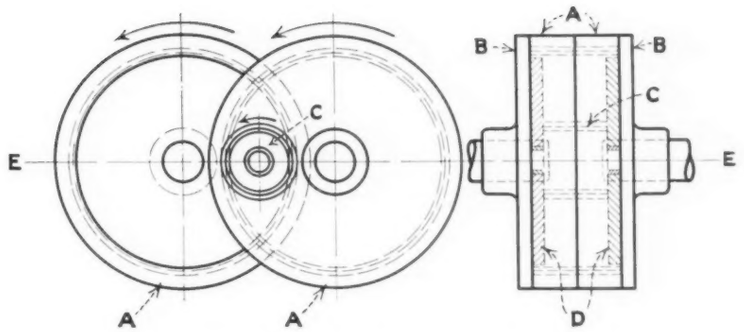
Fig. 3—With the tooth form drawn, the Lewis form factor is determined from the dimension z

the tooth factor for the Lewis equation calculated from the dimension z, shown in Fig. 3. To find the dimension z, the point b is located by cut-and-try, making the distance bk equal to kc, and ca equal to ac. The line bd is drawn perpendicular to ba, establishing the distance z. The tooth-form

Coordinate Coefficients for Laying Out Tooth Forms

Deg.	x	y
5	0.000 221	0.003 80
10	0.001 77	0.015 1
15	0.005 9	0.033 7
20	0.014 0	0.059
25	0.027 2	0.091
30	0.046 6	0.128
35	0.073	0.170
40	0.108	0.215

factor is $2z/3$ divided by the circular pitch, or $z/15.708$ when a diametral pitch of 0.30 is used in the tooth-form generator.



Internal pinion design provides a convenient drive for two short-center shafts revolving in the same direction

Short-Center Gear Drives

To the Editor:

● Special involute tooth forms described by W. A. Tuplin in the March number, page 108, are not the only solution for a drive for short-center parallel shafts rotating in the same direction. A similar gearing condition was met in a special machine for winding the insulation on silk-covered copper wire. In this application, the speed was too high for a chain drive on close center distances.

Two internal spur gears of equal diametrical pitch and pitch diameters, shown at A in the accompanying figure, were closed by the outside covers B. An idler pinion C meshes with the gears and was supported by bushings held by two circular disks D. These disks cleared the internal teeth of the gears and were provided with a running fit on extensions of the two parallel shafts.

Since the directions of rotation for the two gears are opposite the tooth loads on the pinion C will hold it in a balanced position on the center line E. This transmission would operate more smoothly by substituting helical or herringbone teeth.

—C. W. HINMAN
Orlando, Fla.

Visual Properties of Sodium Light

To the Editor:

● Statements given in the short article, "Sodium Light Reduces Eye Strain," page 113 of the March number, challenge discussion of the nature of eye strain.

Incandescent light and electronic, or vapor light, are radically different in their physical nature, and consequently in their visual effect. Incandescent light has a continuous spectrum, while electronic light has the characteristic band spectrum of the element or elements emitting the light. At the ordinary operating temperature of a tungsten light, the light is of a yellow or orange-yellow color, although blue and violet are also present, but not in sufficient quantity to make the light "white."

Electronic light, having a band spectrum, is also colored. The first light of this type to come into commercial use was the mercury-vapor lamp, having five colors: golden yellow, greenish yellow, blue-green, blue, and violet. The two bands of yellow produce over 99 per cent of the brightness effect of the light.

Another type of electronic light produced from neon contains a large number of narrow bands in the red, orange and yellow, although the visual effect is red.

Light from the new form of electronic lamp using sodium vapor as the radiant contains but one color, a double band in the yellow. Since yellow produces the highest degree of brightness for a given amount of radiant energy, the sodium lamp is highly efficient. But this is the sole virtue of the sodium lamp. Its monochromatic yellow light is extremely glaring, notwithstanding the statement given in the article referred to. Instead of eliminating eye strain, it produces eye strain to an intolerable degree when used at the higher intensities required for close visual work.

At the higher intensities, the only monochromatic light that the eye can tolerate is green, provided it is free from any blue component. This is a logical result of the evolution of the eye under conditions in which the green of vegetation is a large element.

The assumption that eye strain is eliminated by the increased visual acuity arising from the elimination of chromatic aberration is without factual proof. The eye focuses for the brightest color at the general intensity of illumination at the time. This color varies from yellow at the highest intensity to greenish-blue or blue at the lowest. The focusing action is entirely automatic, action taking place with a minimum expenditure of energy.

Furthermore, red and blue color fringes resulting from chromatic aberration actually assist the eye in showing the relative positions of objects in space beyond the near foreground. This effect is actually helpful to visual perception, and can hardly be a cause of strain. The fact that yellow light gives the eye abnormal visual acuity does not prove that it produces no eye strain.

With the introduction of new sources of electronic light the illuminating engineer will have a wider choice for specific purposes. But in any application the first question to be considered is the kind of light to be used. Sodium light undoubtedly has a large field, covering exterior lighting and large interiors. The streets and the heavy industries are its natural patrons. In all these instances the intensities are relatively low, and the eye does not work at short focus; so that the glare would not be trouble-

some. In this large field its greatly enhanced efficiency will permit of much higher intensities than are now commonly used, which will be a decided gain. —E. L. ELLIOTT
Formerly Editor, Illuminating Engineer

To the Editor:

● From the personal use I have made of the sodium lamp, the high acuity has not been accompanied by any undue eyestrain arising from the energy distribution in the light spectrum. A number of people, like myself, have worked with these lamps under conditions that are extremely bad from an illumination engineering standpoint, but no case of eye trouble has appeared. From this experience I feel certain that the sodium arc, with its greater acuity, combined with the fact that its color is at the point of maximum sensitivity of the eye, makes the sodium lamp extremely useful under conditions where fine detail must be observed. —FRANK BENFORD
Research Laboratory
General Electric Company

Efficiency of Draftsmen

To the Editor:

● Because of variations in the work produced, standard rates are difficult to establish for drafting, but there is one type of bonus system, the Rowan Plan, that is well adapted to this class of work. This system is based on an hourly rate and allows the draftsman to earn a bonus depending on the amount of time saved over a standard allowable time that has been set for the job.

The bonus is determined by multiplying the money value of the time actually taken to do the job by the time saved and dividing by the standard time. For example, if the standard time for a given job is ten hours, and a draftsman receiving 90 cents an hour does the work in eight hours, his pay for the job will be $7.20 to which will be added a premium of:

$$\frac{7.20 \times 2}{10} \text{ or } \$1.44$$

Even if at times the supervisor makes considerable error in setting the time allowance this system will be an advantage. Since no two jobs are exactly alike, no records of previous jobs need be kept. The supervisor is allowed sufficient latitude in setting time to allow for both the slow men and the fast men.

Where nearly every job is different from the previous one, each new job requires a new estimate, and there is little cause for the draftsman to question the amount of time set for a given drawing.

While it is true that the average drafting room supervisor has a fair idea of what his men can turn out, where there is no system at all, there is certain to be laxity which may represent an appreciable loss in a department of ten or twenty men.

—M. G. DEMOUGEOT
Assistant Factory Manager
International Banding Machine Co.

What the Customer Buys

To the Editor:

● One of the toughest problems the designer faces is that of incorporating into a product exactly those features that the potential customer wants. The General Motors investigation representing the views of 211,000 people may take some guess work out of automotive design. Furthermore, it offers concrete evidence of the popular preferences which merit the attention of all engineers who create products for the non-industrial market.

A similar guide for the designer of industrial and tractor engines is found in a list of requirements given in a paper read before the recent meeting of the Society of Automotive Engineers. The preferences of the public as revealed by the General Motors investigation and requirements for industrial engines as outlined before the automotive society are given here for comparison:

General Motors	*Industrial Engines*
1. Reliability	1. Reliability
2. Operating Cost	2. First Cost
3. Safety	3. Operating Cost
4. Appearance	4. Accessibility
5. Comfort	5. Performance
6. Control	6. Adaptability
7. Smoothness	7. Control
8. First Cost	8. Flexibility
9. Performance	9. Protection
10. Speed	10. Speed Governing

Careful comparison of these two lists leads to interesting observations. Reliability stands first in both columns, operating cost is second in one and third in the other, first cost comes second in the one and places in the other. Other items which differ in their specific meaning but which in general cover safety, convenience and comfort are included in both lists. Performance and control also are in both columns. Appearance, of course, rates well up in the preference of motorists, while flexibility means much in industrial equipment. There is only one feature on which popular and industrial opinion is at loggerheads. That is speed. One requires plenty of it, but the other asks that it be held within a sane limit by governors.

Excellent though it be, the analysis of what the public wants in a motor car should be considered critically when it comes to placing the features in order of importance. While the 211,000 customers may truthfully have told General Motors what they want, they did not necessarily say what they would buy. And there is a difference. The list of requirements given here is one arrived at by reasoning and may be said to stress those features which appeal to the mind. Hence, the high rating of reliability, operating cost and safety. When it comes to actually buying the car, however, the features appealing to the physical senses are likely to play the important rôle, being the stronger motivating force. Therefore, appearance, performance and comfort probably will rate higher under actual conditions of buying. And first cost will not be so far down the line, for the act of paying over cash in sizable quantities is painful to the great majority, if not impossible. —F. K. LAWLER
Assistant Editor, Food Industries

Stresses in a Snap Ring

To the Editor:

● Dimensions of the snap-ring design requested by E. P. Hatchel in the February number on page 70, can be readily obtained by the following mathematical analysis. Considering *dl* as an element of a bent beam, as shown in Fig. 1, the load *P* at point *k* introduces a moment in *dl*, causing the angular deflection *da* given in the

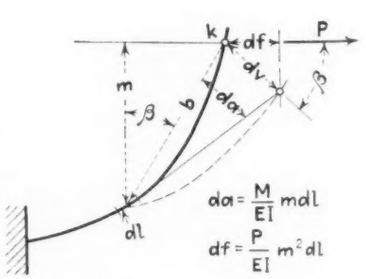


Fig. 1—Stresses in a snap ring can be computed by a formula developed from equations for the deflections of an element

first equation in Fig. 1. When point k moves through an arc for a distance $b\ d\epsilon$, the projection of this movement df in the direction of P is shown by the second equation. With $b \cos \beta$ equal to m in an elastic beam of modulus E and moment of inertia I, the total deflection F in the direction of P is:

$$F = \frac{P}{EI}\int m^2 dl \qquad (1)$$

For a part of a circular beam as shown in Fig. 2, the lever arm m is $r \cos \epsilon - r \cos \gamma$, and dl is $r\ d\epsilon$.

By inserting these substitutions into Equation (1) and integrating for the full circle angle of 2π, the spreading of a full circular ring of diameter D is given by:

$$F = \frac{3\pi PD^3}{8EI} \qquad (2)$$

Introducing the thickness of the ring cross section, considering that for

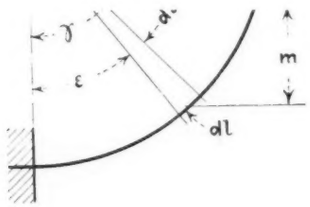


Fig. 2—Integration over the entire circle shows that for a given maximum stress the thickness of the ring varies with the square of the diameter

a rectangular section thickness T is $2y$, Equation (2) becomes the maximum stress formula in the following form:

$$S_{max} = PDT/2I \qquad (3)$$

Eliminating P/I and combining Equations (2) and (3), the thickness T in terms of the diameter, maximum stress, and deflection is given by:

$$T = \frac{3\pi\ D^2 S_{max}}{4EF} \qquad (4)$$

The width of the ring does not enter into the calculations. For all practical purposes P can be figured from this formula if the diameter of the deflected ring is considered.

In the problem stated by Mr. Hatchel, this final relation gives a thickness of 0.108 in. for T. From Equation (3) the spreading load P is 8.2 lb. By combining Equations (2) and (3) it can be shown that for a given application P increases with the third power of S.

—Walter Schroeder
Cincinnati, Ohio

Shear Pins

To the Editor:

● In the article "Design Formula for a Pin and Shaft of Equal Strength," appearing on page 187 of the May number, the analysis given is complete except for compression stresses in the hole. After the pin diameter has been calculated, a careful check should be made on the compression or bearing stresses in the shaft hole otherwise distortion of the hole may progress so rapidly that the shaft is damaged, thus defeating the whole purpose of the safety shear pin.

—T. J. Bower
New York, N. Y.

Errata

● In the Reference Book Sheet appearing on page 160 in the April number, the minus sign in the Barth factor equation should be plus. The correct form for this equation is:

$$A = \frac{a}{a + V}$$

Equation (3) in the article "Jewel Bearings for Minimum Friction," appearing on page 179 of the May number, should include the factor $(E_1 + E_2)$ in the numerator instead of the denominator. The correct form is:

$$R = \frac{16\ (E_1 + E_2)\ \sqrt{3\ P}}{(2\pi\ k)^{1.5}}$$

Barth Factors for Worm Gears

To the Editor:

● When wear is the principal consideration in the design of worm gearing, the strength, generally, will be in excess of requirements. And when shock loads are not present, static loadings do not need to be reduced by the use of the Barth equation as given in Mr. Grill's article in the February number of *Product Engineering*.

Considering that the wearing surfaces in worm gears are comparable to the corresponding elements in a plain bearing, the static loadings can be increased in proportion to a derivative of the velocity up to certain limits, possibly 300 ft. per min. Running loads will be less because of more efficient lubrication and lower coefficients of friction. Experiments conducted by Wilfred Lewis for William Sellers & Company showed this increase of efficiency with increased speeds of rubbing between gear and worm.

As shown in the April number, page 160, the Barth equation for velocity factor has two commonly used constants, 600 and 1200, the factor 600 being used for commercial low-speed drives and 1200 for gearing of more accurate manufacture. Most worm gears are manufactured on hobbing machines which can produce a very accurate surface. For this reason worm gears subjected to shock loads should be designed using the factor 1200 in the Barth equation, allowing higher working loads than recommended by Mr. Grill.

When the rim on worm gears is designed with a straight face, the gear blank can be machined as though it was for an equal spur gear. This form makes it possible to turn the blank accurately, and allows accurate setting of the hob to the outside diameter when gaging the depth of cut.

A worm gear of straight face has a projected tooth area equal to that of the old style gear face made with the sides radial to the worm center. For the same overall face width, the form given by Mr. Grill will show a slightly larger area, but this extra area is beyond the outside diameter of the throat. Since this area moves in an interference field, it undoubtedly does not carry its proportionate load. For this reason, this added area is seldom worth the extra cost of production.

—P. N. Bland
Vancouver, Canada

To the Editor:

● To produce accurate gears by hobbing requires precision hobs and the use of considerable care in the cutting operation. For loads as high as recommended in the article on page 2 of the January number, commercial gear accuracy is not sufficient. Therefore, unless the designer knows the kind of a gear he is using, the method outlined in this article, and Mr. Bland's suggestions as well, should be used with caution.

Although the straight face worm gear has the advantages outlined by Mr. Bland, the actual number of teeth in contact is less than when the curved face gear is used. For this reason, the curved face should be given preference unless it can be shown that the straight face worm gear is equal to it in carrying capacity and gear life.

Straight-face worm gears, popular some years ago, have been abandoned for reasons unknown. Perhaps "handbook designing" was responsible for the disappearance of this type along with bevelled edge worm gears.

—C. H. Grill
Highland Park, N. J.

Hinges and Pivots for

HENRY J. MARTIN

Fig. 1—Common cover hinge with pin tight in the cover and loose in the hinge lugs.

Fig. 2—With the end peened the pin can be made a loose fit in all lugs.

Fig. 3—A plain pin with two cotters can be used in place of a peened rivet.

Fig. 4—A double tapered hole in cover lugs with pin fitting tightly in outer lugs.

Fig. 5—Sheet metal cover bent around the hinge pin.

Fig. 6—A tapered pin makes hinge adjustable.

Fig. 7—Combination straight and taper pin hinge.

Fig. 8—Hardened and ground washers to prevent wear. **A**, hinge lugs milled to prevent washers from turning. **B**, washers retained by a pin.

Fig. 9—Pivot bearing with shouldered center pin and adjustable cone pointed screw with lock provided.

Fig. 10—A design for severe duty and long life.

Fig. 11—Capped eccentric pin bearing makes cover adjustable.

Fig. 12—Pivot bearing with hardened conical pointed pins.

Fig. 13—Sheet iron cover swinging on headed pin which is peened over at the opposite end.

Fig. 14—Lever arm pivoted on shouldered stud and retained by washer.

Fig. 15—Light cast iron cover pivoting on stud.

Fig. 16—For heavier covers, the stud is shouldered and the casting counterbored.

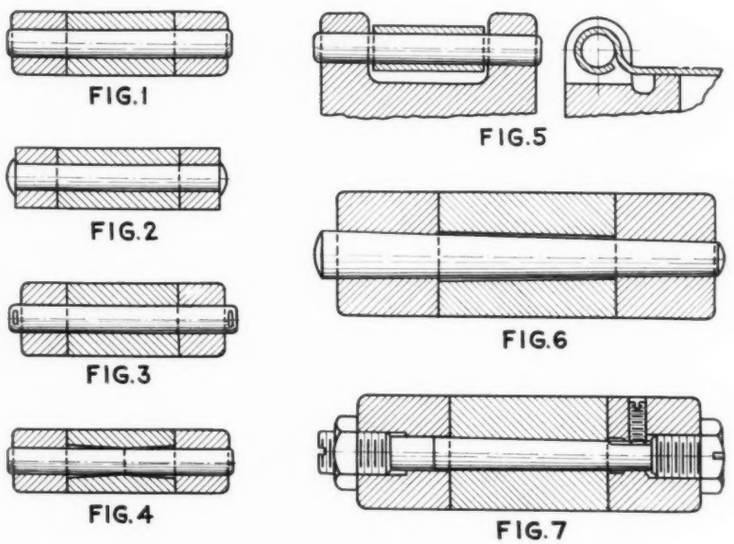


FIG.1 FIG.2 FIG.3 FIG.4 FIG.5 FIG.6 FIG.7 FIG.8

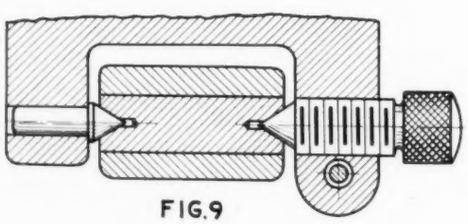


FIG.9

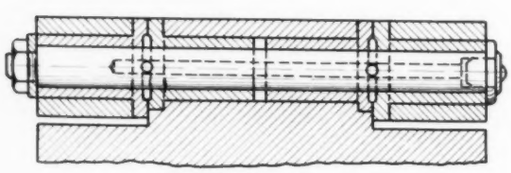


FIG.10

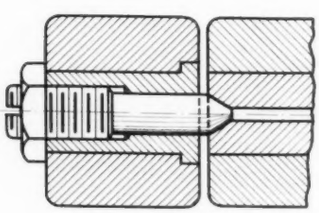


FIG.12

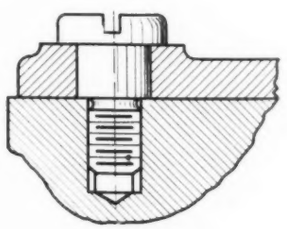
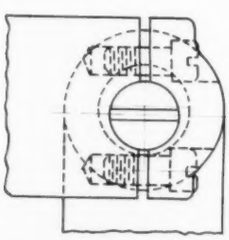


FIG.15



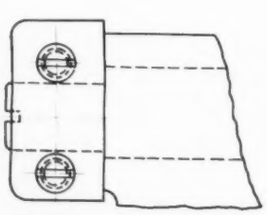


FIG.11

FIG.13

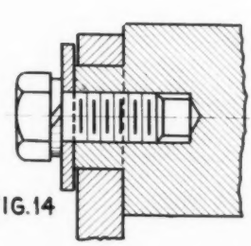


FIG.14

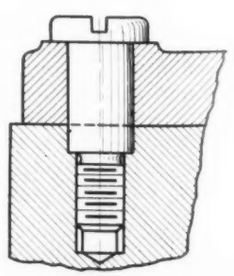


FIG.16

Covers and Flexible Joints—I

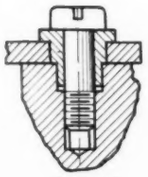


FIG.17

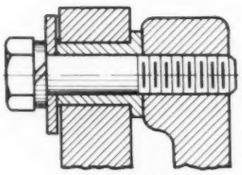


FIG.18

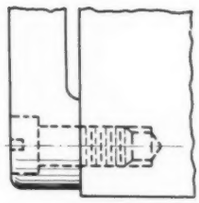


FIG.20

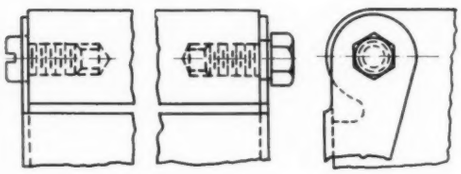


FIG.19

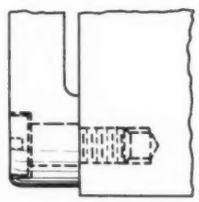
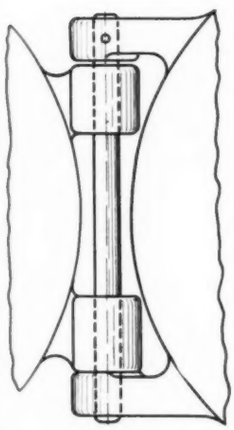


FIG.21



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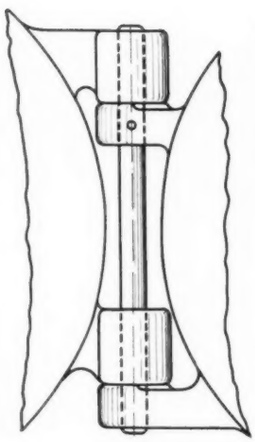


FIG.23

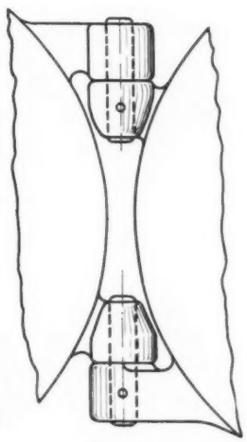


FIG 24

Fig. 17—Shouldered bushing centered in counterbored hole.

Fig. 18—Bushing centered and held by stud screw.

Fig. 19—Sheet iron cover with hinge ears fastened with shouldered hex head stud, or filister head screws.

Fig. 20—Poor design because screw tends to turn.

Fig. 21—The hinge stud should screw in tightly and head should have ample clearance.

Fig. 22—A poor design wherein all the thrust is taken on the lower case lug and the span of the cover lug is reduced.

Fig. 23—A conventional machine door hinge.

Fig. 24—An improved design with lug drilled and reamed from opposite ends.

Fig. 25—An incorrect design because it requires removal of the small pin for disassembling.

Fig. 26—With a headless hinge pin, the cover can be lifted off.

Fig. 27—Separable lugs are used when the casting is too large for small lugs to cast satisfactorily.

Fig. 28—Another method of holding the steel lug.

Fig. 29—Wrong method of fastening steel lug, requiring cut and try. End view shows results.

Fig. 30—With inverted pin, the cover lug can be smaller. Studs are positioned before the final pinning.

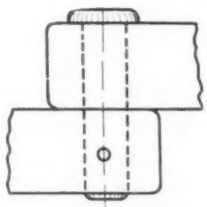


FIG.25

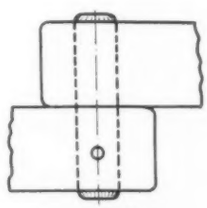


FIG.26

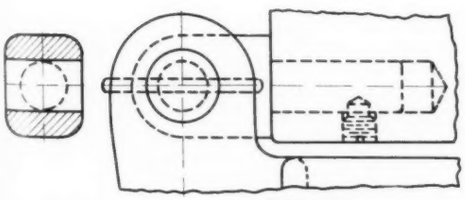
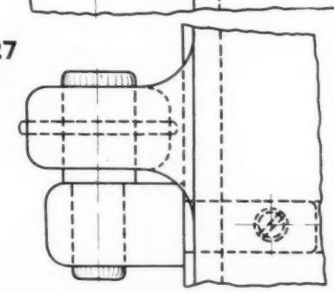


FIG.27



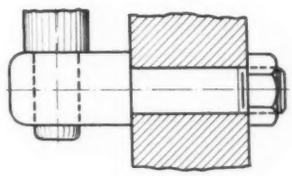


FIG.28

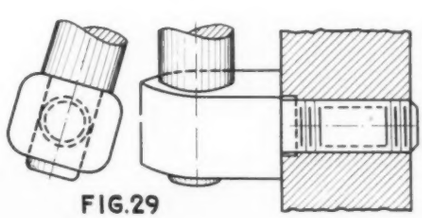


FIG.29

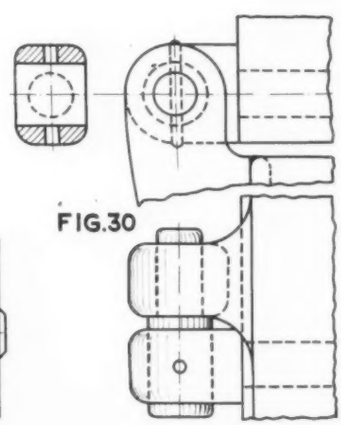


FIG.30

· NEWS ·

Less Friction With Alumilite Piston

ANODIC coatings on automobile pistons reduce friction and scoring in cold starts, according to Austin Wolf, consulting engineer, in a talk May 17 before the Metropolitan Section of the Society of Automotive Engineers. Piston coatings are an important factor in starting in cold weather, since piston and ring friction alone account for more than half the friction loss in the engine. Under the extreme weather conditions experienced last winter, new cars with close clearances may not develop sufficient power from initial explosions to overcome the friction load. By reducing the piston friction, anodic coatings on the pistons make cold starts possible with less wear on the cylinder wall, as well as imposing a smaller load on the electrical system.

A.S.T.M. to Report New Corrosion Data

PROPOSED tentative specifications on stainless steels, and aluminum, zinc, and magnesium alloys, and detailed reports on the atmospheric corrosion tests occupy a prominent place in the committee meetings and discussions scheduled for the 1934 Annual Meeting of the American Society for Testing Materials to be held June 25-29, in Atlantic City.

Tentative specifications will be proposed for 18 per cent chromium-8 per cent nickel steel. These specifications will cover the application of welded 18-8 stainless steel used at ordinary temperatures for corrosion resistance. Chemical, physical, mechanical and fabricating properties of this material are scheduled for discussion.

Atmospheric tests on galvanized sheets after seven years' exposure and metallic-coated hardware and structural materials exposed to weather at several test locations will also be reported at this meeting, along with other tests of electrolytic corrosion in groups of specimens in 9 outdoor locations. Results of specimens subjected to total immersion tests will also be described.

Corrosion resistance is likewise an important factor in the proposed tentative specifications and reports on aluminum-base and zinc-base die castings, and magnesium-base alloys. Discussions of the zinc-base alloys will also include data on the range of impact values for several types of cross sections as compared with standard test specimens.

Coated Electrodes for High-Quality Welds

WHILE weld metal and welded joints made with bare or lightly coated electrodes are usually satisfactory for static loads, they have relatively low ductility, impact strength, fatigue strength and corrosion resistance, as explained by J. E. Waugh of the Welding Engineering Department of the General Electric Company in speaking before the International Business Machines Club at Binghamton, N. Y. The speaker illustrated how the heavily coated electrodes have made it possible to produce welds having properties comparable to good-quality, mild-steel plates, and presented the accompanying table, giving the comparative values obtained with the two types of electrodes, bare and heavily coated.

As explained by Mr. Waugh, the superior physical properties of the welds made from heavily coated electrodes are developed by shielding the molten metal while it passes through the arc, thus eliminating the oxides and nitrides which have a great affinity for molten iron at the arc temperature. Further improvement in the physical property of the weld is brought about by the refining effect of each succeeding pass on the preceding pass, when more than one bead is welded. For welds that are to have maximum ductility, the completed weld should be annealed at 1,200 deg. F., to relieve strain.

According to the speaker, there is a decided tendency towards replacing bare electrodes with heavily coated electrodes, in spite of the cost differential of the electrode material, which is in the ratio of two or three to one in favor of the bare electrode. In addition to this, the heavily coated electrode is more expensive to apply, largely because of the time involved in removing the slag between passes and the necessity of training welders in the use of heavily coated electrodes. However, the superior qualities of the welds made from heavily coated electrodes offsets, in many instances, their somewhat greater cost.

Diesel Requirements for Tractors

IN discussing diesel tractor problems, H. H. Howard of the Caterpillar Tractor Co., before a recent meeting of the Society of Automotive Engineers, stated that, "No diesel tractor business in reasonable volume may be expected unless the engine will render service as generally satisfactory as the gasoline engine. Operators will not tolerate any unusual maintenance or operating difficulties."

Characteristics essential in the design of the tractor diesel to meet these

Comparative Physical Properties of Welds and Welded Joints made with Bare and Heavily Coated Electrodes

WELDED JOINT

	Bare Electrodes	Heavily Coated Electrodes
Tensile Strength in lb. per sq.in.	50-55,000	Fracture in plate
Per cent elongation in free-bend test	10-15	30-50

WELD METAL

	Bare Electrodes	Heavily Coated Electrodes
Yield point in lb. per sq.in.	40-45,000	50-55,000
Ultimate tensile strength in lb. per sq.in.	50-55,000	60-70,000
Per cent elongation in 2 in.	5-10	20-30
Per cent reduction of area	10-20	35-50
Charpy impact in ft.-lb.	2-5	20-30
Specific gravity	7.70	7.82
Endurance limit in millions of cycles	10-15	20-30
Corrosion—(mild-steel plate as 100)	50	95-100

requirements are given by Mr. Howard as follows:

1. Positive starting
2. Ability to follow changes in load
3. No pounding or roughness
4. Maintenance no greater than for gasoline engines
5. Servicing by the tractor operator
6. Filtered air and fuel supply
7. No critical service adjustments
8. Operation on a wide range of fuels

For this service, light weight, noise reduction, instant starting and high speed are not required, although for automobile or truck service such characteristics would be important.

Practical Uses of Color

COLOR can serve many useful purposes in addition to being an adjunct of usage, according to the Duco Color Advisory Service. Thus, red is used to accentuate the part to which it is applied, while colors of the order of blue and purple have merit in fixing the relationship of various parts that are to appear distant. When it is desired to increase the appearance of heights or size, white or light bright colors are used. Black or dark colors prove effectual in creating an illusion of solidarity, compactness or lowness.

Because they do not suggest "business," colored typewriters have found favorable acceptance in the home. White rice and white eggs are marketed in blue-lined containers in order to increase the white of the product by contrast. Our eyes are best accustomed to yellow-green light, consequently it is by this light that we see best.

Psychologically, black and gray are conducive to melancholia and general depression. Red induces temporary stimulation. Blue induces calm and contentment. Brown can impart an atmosphere of restfulness or depression, according to the amount used.

In marine engineering it has been found that colors of high luminosity have a marked effect in decreasing the rate at which barnacles accumulate on the submerged portions of boats.

Dark places require strong, bright contrasting colors. Strong chroma yellows, yellow-reds, reds, greens and blues, together with white, are readily discernible and attractive in combination at dusk. The reflection factor of white is 88 per cent, which is 67 per cent more efficient than a tint of yellow and 78 per cent more efficient than a tint of green.

Large gasoline tanks are painted white to avoid excessive evaporation. A five section radiator painted white radiates as much heat as a six section radiator painted in aluminum, according to U. S. Bureau of Standards tests. As many as fourteen coats of paint were used on one radiator but only the last coat influenced the heat radiation.

Research With Models

A NATIONAL conference and exhibit on the use of models in research and technical education will be held in Ithaca, New York, June 19-21, in connection with the annual meeting of the Society for the Promotion of Engineering Education. Prof. Fred L. Plummer of the Case School of Applied Science will be general chairman. Papers will be presented on models relating to problems in both hydraulics and mechanics. The conference will bring together from all over the country men who are doing pioneer work in the fields of materials, hydraulics and structures.

On the afternoon of Tuesday, June 19, Hale Sutherland of Lehigh, T. H. Evans of Sheffield Scientific School and H. J. Gilkey of Iowa State will present papers on "The Fundamental Principles of the Deformeter Method of Model Analysis," "Photoelasticity—a Short Explanation of the Optical Principles Involved," and "The Various Types of Loaded Models and Some Problems to Which Their Use is Adapted." Papers to be presented Wednesday, June 20, and of special interest to designing engineers, include, "Applications of Loaded Models," by J. C. Rathbun of City College; "Photoelasticity," by M. M. Frocht of Carnegie Institute of Technology; "Deformeter Method of Model Analysis," by J. T. Thompson of Johns Hopkins University; "Research Employing Photoelastic Methods," by G. B. Karelitz of Columbia University; and "Soap Film Analogy," by Bruce Johnson of Lehigh University.

In addition to those mentioned, there will be fourteen papers on the application of models in problems relating to hydraulics and civil engineering structures.

In general, there will be scheduled discussions of each paper. General discussion will be allowed insofar as possible. Exhibits and demonstrations will be on display throughout the conference. Special "small group" luncheons will be held on Tuesday and Wednesday to allow speakers and others to discuss informally some of the major model studies which are now under way or completed. The Annual Meeting of S.P.E.E. continues from Thursday noon, June 21, until the following Saturday noon. The conference is open to all interested and no registration fees are charged.

MEETINGS

Society of Automotive Engineers—Summer meeting, Saranac Inn, Lake Saranac, N. Y., July 17-22. John A. C. Warner, secretary, 29 West 39th St., New York, N. Y.

American Institute of Electrical Engineers — Summer meeting, Homestead Hotel, Hot Springs, Va., June 25-29. H. H. Henline, secretary, 33 West 39th St., New York, N. Y.

American Society of Mechanical Engineers — Semi-annual meeting, Cosmopolitan Hotel, Denver, Colo., June 25-29. Calvin W. Rice, secretary, 29 West 39th St., New York, N. Y.

American Society for Testing Materials—Annual meeting, Chalfonte-Haddon Hall, Atlantic City, N. J., June 25-29. R. E. Hess, assistant secretary, 260 S. Broad St., Philadelphia, Pa.

A.S.M.E. Holding Session on New Trains

AT THE semi-annual meeting of the American Society of Mechanical Engineers to be held in Denver, June 25-29, a session is scheduled for a discussion of the new high-speed, light-weight trains.

Papers on the performance of the new railroad trains will be given by representatives of both the Burlington and the Union Pacific Lines. Service experience by both of these operators should provide data of considerable interest for this meeting.

The A.S.M.E. also announces that these light-weight trains are expected to be available for inspection at Denver during the meeting.

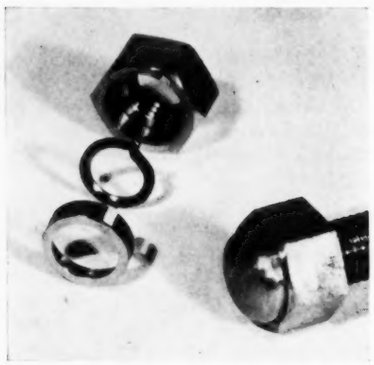
New Name for the Bureau of Standards

NATIONAL Bureau of Standards now replaces U. S. Department of Commerce, Bureau of Standards, as the official designation of that body. The new name restores the original title, the purpose being to eliminate confusion with the symbol "BS" used by the British Standards Institute. Under the new name "NBS" designates the National Bureau of Standards.

NEW MATERIALS and PARTS

Safety Nuts

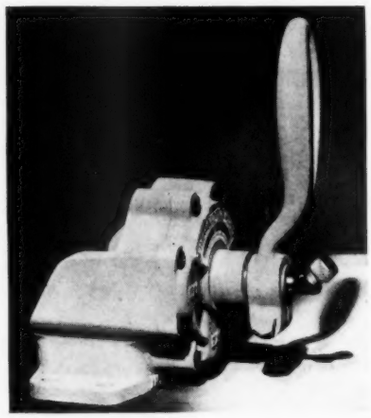
Consisting of three parts; a nut of same dimensions as a standard castellated nut, a spring ring of triangular section, and a closing cap. Top of nut is counterbored approximately two threads, and the spring ring is inserted with the extension fitting in the slot in the nut. Locking ring has a triangular section of the same dimensions as the thread. Made of cold-drawn 0.50 carbon spring wire, not hardened, spring lock is coiled so that it offers little resistance when the nut is screwed on the bolt. In unscrewing the nut, wrapping action causes the locking ring to clamp the thread, offering high resistance. According to the manufacturer's claims, the lock nut can be attached and removed repeatedly without affecting the locking action. Counterbore in the nut extends to a depth equivalent to the added height of a castellated nut,



leaving the same number of active threads as used in standard nuts. Available in all standard sizes, closed as shown at the right to prevent entrance of dirt and water, or open as shown at the left. Safety Nut Corp., Land Title Bldg., Philadelphia, Pa.

Hanna Manifold Valve

Designed for installation where a number of valves must be located close together, piping can be arranged without interference with adjoining valves. Either intake or exhaust opening may be connected to a manifold. For air, oil or water, and pressures to 250 lb. per sq. in., full port opening is obtained with a 40 deg. movement of the



operating handle; a reversal with an 80 deg. movement. Two-piece disk and stem rotate as a unit, lubricated from a well in the stem. Scoring is said to be prevented by rotating the disk on two eccentric points, distributing wear across the full diameter. Valve disk can be removed without disturbing pipe connection or removing the valve body from the pipe line. Hanna Engineering Works, 1769 Elston Ave., Chicago, Ill.

Chase Olympic Bronze

Available in three types: Type A in hot-rolled and cold-rolled forms; Type B in free turning wire and rod; Type C for castings. Type A is said to be readily cold-rolled to 75 per cent reduction in thickness without becoming brittle. Is forged at a temperature of 1,650 deg. F., flowing 70 to 90 per cent as much as brass at 1,450 deg. F. Machinability of Type A said to be slightly better than hard copper, while Type B is machined in automatic screw machines at $\frac{3}{3}$ the cutting speed used for free cutting brass rod. Type C is recommended for high strength sand castings, particularly for fine detail work. Approximate strengths in various forms are said to be:

Form	Tensile Strength Lb. per sq. in.
Hard drawn wire	150,000
Hard drawn sheet	115,000
Bolt stock	90,000
Forgings	60,000
Sand Castings	40,000

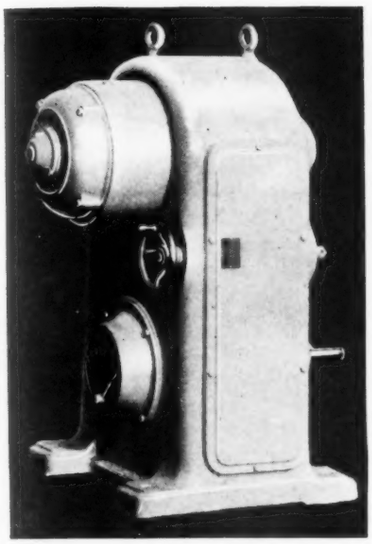
Type A available in strip, sheet or plate in any temper. Type A and B both available in rod and wire sizes from 0.020 to 4 in. in diameter in any temper. Type C supplied in form of castings or ingots. Chase Brass & Copper Co., Inc., Waterbury, Conn.

Delta-Star Receptacle

For quick disconnection of control wiring, receptacle can be bolted flat against an opening in a switch housing, inclosing the control wiring within the receptacle. The type illustrated is a seven pole 60 amp., 600 volt receptacle, the rating being for continuous service. Also available in types having 2 to 6 poles. Receptacle cannot be used for opening or closing circuits under load. Cover automatically closes when the plug is removed. Delta-Star Electric Co., 2400 Fulton St., Chicago, Ill.

Sterling Upright Vari-Speed Motors

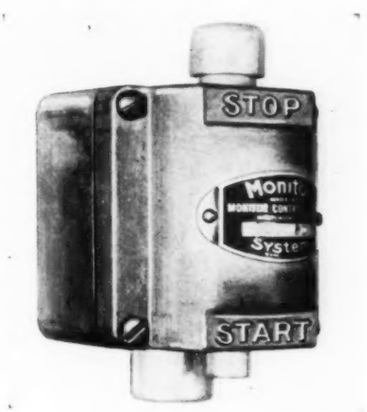
Incorporate in a single vertical unit, an electric motor, speed reducer with infinite variation in ratio, and a speed-ratio indicator. The 10 hp. design illustrated has a speed range from 1,200 to 300 r.p.m. Drive is by a flexible V-belt running on two V-pulleys mounted on parallel splined shafts. Hand wheel simultaneously changes the diameter of both pulleys, changing the drive ratio without altering belt tension. Ball bearings are used throughout. Design is said to provide constant speed at any setting. All



parts are rustproof. Available in ratings up to 50 hp. with standard speed variation of 2:1, 3:1 and 4:1. Also available with remote control, higher ratios and double reductions. Sterling Electric Motors, Inc., Los Angeles, Calif.

Monitor Weatherproof Control Station

As shown in the illustration, stop button is conspicuously placed for accessibility, while the start button is protected to prevent accidental operation. Fixed contacts are mounted on a square bakelite plate, fastened to the base with one screw. Contact plate can be adjusted to conform to the position of the conduit entrance, which may be turned to top, bottom or either



side. Contacts are set for operation with the stop button at the top. Available in single, two or three button types, with a variety of contact arrangements for machine tool applications and other pilot circuit controls for magnetic motor starters. Monitor Controller Co., Baltimore, Md.

"Thiokol" Sheathed Electrical Cord

Rubber insulated and claimed to be oil proof. The cord is sheathed with "Thiokol"—a rubber-like material impervious to oil and gasoline—to protect the rubber insulation against rotting or becoming swollen and sticky. General Cable Corp., 420 Lexington Ave., New York, N. Y.

Haskelite Plymetl

A new type of wall panel consisting of ply-wood with a bonded metal face. For interior use, only the exposed face is covered using copper or galvanized steel. Also available with polished or satin finished chromium alloy. For outdoor construction, the finished surface is bent over the edges and soldered to a galvanized steel back attached to

the other side of the ply-wood sheet. Strength is said to be many times that of sheet steel, depending on the thickness of the ply-wood core and the face metal. Panels can be bent to a radius as small as 7 in. Available in stock sizes up to 36 in. x 96 in. Haskelite Mfg. Corp., 208 West Washington St., Chicago, Ill.

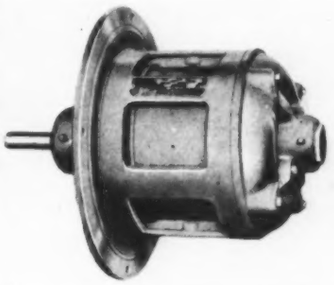
Champion Plain End Pipe Couplings

Designed to connect plain end pipe without grooving, upsetting, threading or beveling the ends of the pipe. Connection is said to provide strength in tension sufficient to withstand strains encountered during connecting and while the line is under pressure. Coupling consists of five parts, two half-couplings which contain the gripping element, a gasket and two bolts. Couplings and gripping rings are drop forged and heat-treated with gripping rings assembled permanently into the half-coupling. Gaskets provide a double seal which is said to be equally effective against either pressure or vacuum. Steel reinforcements protect rubber gasket from pinching action on the pipe and are also said to prevent blowing out of the gasket between the

lugs of the coupling when applied to pipe whose diameter tolerance is on the large side. The joints can be taken apart and reassembled any number of times without impairing the efficiency of the coupling, according to the manufacturer. Available for all standard pipe sizes. The Champion Machine & Forging Co., 3695 East 78th St., Cleveland, Ohio.

Flanged Head Motors

Standard electric motors with flanged head mountings applied as an integral part of the motor head. Flange may be finished on either or both sides. For use on close-coupled pump mountings, machine tools, including lathes, grinders, polishers, etc., and other applications where space limitations make it desirable to eliminate motor feet and provide finished flange on motor head for end mounting. The Louis Allis Co., Milwaukee, Wis.



Electrode for Class I Welds

Designated as type W-22, a heavily coated or shielded-arc type is designed to produce welds of the quality required for Class I pressure vessels in accordance with the A.S.M.E. Boiler Construction Code. Electrodes can be

used in any position for flat, vertical or overhead welding, and at the same time are said to have deep penetrating properties, being suitable for either butt or fillet welds. Average results from tests of specimens consisting entirely of weld metal are:

Tensile strength
65,000 to 75,000 lb. per sq.in.
Elongation .....20 to 30 per cent in 2 in.
Impact resistance.30 to 45 ft. lb. (Charpy)

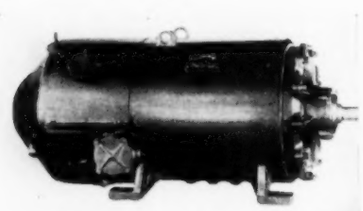
Because these physical properties are obtained without regard to the position in which the welds are made, there is said to be no necessity for moving the work in order to weld in a downward position. Handling costs are said to be reduced in fabrication of large structures, such as ships and locomotives where the work cannot be moved into a special welding position, and in welding fittings to pressure vessels. General Electric Co., Schenectady, N. Y.

High Strength Welding Electrode

For use on high strength steel, a new arc welding electrode is said to develop welds with a tensile strength of 85,000 to 100,000 lb. per sq.in., with a ductility of 15 to 20 per cent elongation in 2 in. Heavily coated electrode is of the shielded-arc type, said to be particularly suitable for welding low alloy structural steels containing less than 0.30 per cent carbon. Welding rod is suitable for flat, vertical and overhead welding, and is available in diameters of $\frac{5}{32}$ and $\frac{3}{16}$ in., the larger size recommended for flat welds and vertical welds in thick plate. The Lincoln Electric Co., Cleveland, Ohio.

Direct-Current Explosive-Proof Motor

For use in Class I Group D applications for dry cleaning, paint and lacquer equipment or in breweries, oil distilleries, or bulk oil stations where only direct current is available. Motor

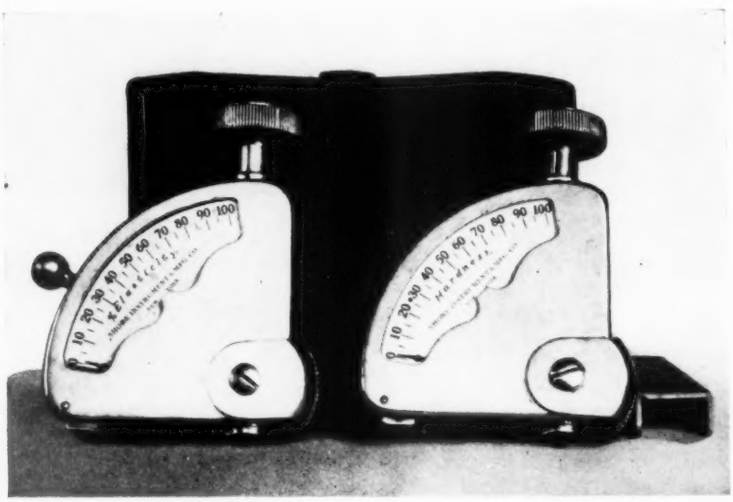


is explosion-proof and fan-cooled. Removable screw caps are provided for inspection of commutator and brushes. The Louis Allis Co., Milwaukee, Wis.

FOR THE ENGINEERING DEPARTMENT

New Shore Durometer and Elastometer

For measuring hardness and elasticity in rubber products such as typewriter rolls, valves, disks, tires and tubes, the Durometer shown on the right is available in five ranges for measuring hardness, and the Elastometer at the left in a single range for measuring elasticity or stretching properties. Hardness measurements are given in terms of unit resistance in kg. per sq.cm. by means of an impresser point extending through the abutment plate in the base of the instrument. Hand pressure applied to the instrument when the abutment plate is placed over the material to be tested, forces the impresser point into the rubber under a fixed spring pressure. Deflection of the impresser point is indicated on the dial. An adjustable guard is provided to protect the point from injury when not in use. Instrument case of hard drawn brass is nickel plated and a leather carrying case is supplied.



For measuring elasticity, the Elastometer is similar to the hardness measuring instrument, but has no resisting spring supporting the penetrator pin. Before the test the pin is locked in the full projecting position. Pressure against the pin is indicated on the dial in units equivalent to one-fourth the reading from a standard stretch test. The two instruments together are said to be necessary to determine the quality of rubber, the product of the hardness reading and the elasticity reading indicating the efficiency of the rubber for general service. The Shore Instrument Mfg. Co., Jamaica, N. Y.

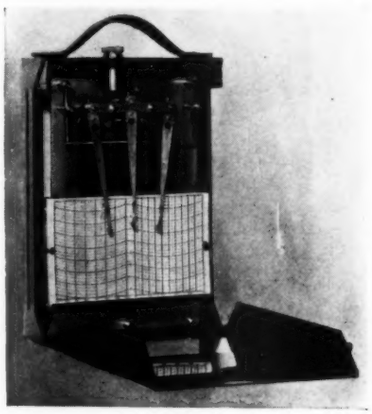
Trofimov Problem Folder

A four-page 9x12 in. folder printed on heavy paper, presenting an orderly procedure for the solution of any problem. The procedure outlined is elementary and highly general, and is said to be applicable to every type of problem, using methods presented by the author in recent numbers of *Product Engineering*. On the first page is provided space for stating the need and a statement of the problem. A diagram of the steps in the procedure is reproduced on page 2, while an explanation of the steps in this procedure are given in detail in the third page. Thirteen suggestions of a general nature are listed on the last page. L. A. Trofimov, 3859 Northampton Road, Cleveland Heights, Ohio.

Friez Recorders

Records humidity, temperature and operation using card charts. As shown in the illustration, the record chart consists of a card 3x5 in., which is

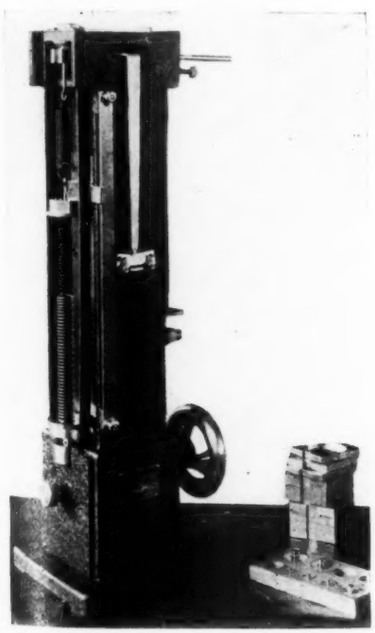
moved downward by a clock mechanism. Time lines are common to all three pens to assist in making accurate comparison between the three records. Die-cast aluminum case is provided with a leather handle and an unbreakable observation window. Case is 8½x5½x2¼ in., weighs 52 oz. Available with three temperature ranges, one humidity range from 0 to 100 per



cent, with operation recorder designed for use in series with loads from 2 to 8 amp. a.c. or d.c. Thirty hour charts are standard with ten hour timing mechanism and charts available. Julien P. Friez & Sons, Inc., Baltimore, Md.

Elasticometer Spring Testing Machine

Provides a rapid and accurate static test for light springs, either tension or compression up to 5 lb. Driven by the hand wheel, a rack carries the upper compression plate, the lower tension hook and the pointer for indicating spring length. The weighing scale consists of a 1:1 beam supporting the lower compression plate and the upper tension hook from one end of the beam. Calibrated weights are suspended from the other end. To reduce lateral friction, the lower compression plate is guided by steel balls, preventing the scale from jamming and permitting free play of the beam in the balance position. Carried on the beam, a pointer magnifies deviation by four times. Rubber bumpers are provided to absorb shocks when springs are suddenly loaded or unloaded. Vertical scale for measuring spring deflections is attached to the stand. Both sides are graduated either in fractions or decimals of an inch, and a vernier permits readings to 0.01 in. Accommodates springs up to 12 in. long. Area of base plate, 6¼x6¼ in., height, 23 in. Also available for loads up to 12½ lb. Coats Machine Tool Co., Inc., 112 West 40th St., New York, N. Y.



MANUFACTURERS' PUBLICATIONS

Ball Bearings—New Departure Manufacturing Co., Bristol, Conn. Two booklets giving design details, bearing layouts, constructional features and applications of sealed ball bearings for mine car wheels and conveyor-roller bearings permanently lubricated and sealed.

Bronze—Chase Brass & Copper Co., Inc., Waterbury, Conn. Booklet, 8½x11, 32 pages, "Chase Olympic Bronze," a well-prepared, complete data book giving fabrication, tensile and other mechanical properties of three grades of bronze. Data on physical and electrical properties and corrosion resistance are included. Available forms, sizes, tempers and weights are arranged in a series of tables.

Chain Drives — Link-Belt Co., Indianapolis, Ind. General Catalog 600, 6x9 in., 1,024 pages. Included with catalog information on chains, sprocket wheels, pulleys and gears are sections giving engineering data, power capacity, weights, useful tables and design information. Pillow blocks, shaft hangers, speed reducers, flexible couplings, elevator buckets and other products are also listed. Typical layouts, design data and types of drives are given in a section on conveyor drives, and a concluding section of 32 pages tabulates engineering constants, weights of materials and other useful data.

Electric Motors — The Louis Allis Co., Milwaukee, Wis. Bulletin 505-C illustrating construction and application of squirrel cage motors.

Electric Motors—Wagner Electric Corp., 6400 Plymouth Ave., St. Louis, Mo. Bulletin 167, Part 8, four pages, giving construction details on RD, direct-current, compound-wound motors.

Friction Clutches — Allis-Chalmers Mfg. Co., Milwaukee, Wis. Catalog No. 2068, 48 pages, 5x7 in., includes data on selection of proper size clutch and catalog description of friction clutches, couplings, clutch operators and friction clutch pulleys.

Gears — Boston Gear Works, Inc., North Quincy, Mass. Catalog, No. 50, 240 pages, 3½x6 in., "Boston Power Transmission Products," superseding previous editions, includes dimensions and ratings for brass, die-cast, fabric, cast iron and steel gears, pinions, worms, bevels and spiral gears. Chain drives, speed reducers, and universal joints are also listed.

Inventions—Inventions Digest, Inc., 310 S. Michigan Ave., Chicago, Ill. Folder, 8 pages, announcing a service to manufacturers reporting progress and availability of new inventions.

Measuring Tools—The Van Keuren Co., 12 Copeland St., Watertown, Boston 72, Mass. Catalog 29, 6x9 in., 20 pages, "Van Keuren Precision Measuring Tools," has catalog information and uses for light-wave equipment, rectangular reference gages, surface plates, light-wave micrometers and flatness testers. Several types of thread measuring devices are described along with suggestions for their use.

Plastics — General Plastics, Inc., North Tonawanda, N. Y. Folder, 8 pages, "Durez Molding Compounds," prepared for the designer includes data on advantages, limitations and properties.

Plastics—Toledo Synthetic Products, Inc., 2124 Sylvan Ave., Chicago, Ill. Booklet, 8½x11 in., 20 pages, "Plaskon Molded Color," attractively illustrates in full color a number of hot-molded products made from a clear powder colored as desired before molding.

Stainless Steel — Republic Steel Corp., Massillon, Ohio. Booklet, 60 pages, 8½x11 in. "Enduro 18-8" includes data on typical analyses, physical properties and fabricating processes. Physical properties at high temperature, and laboratory corrosion data are given in complete form.

BOOKS and RESEARCH BULLETINS

Thermostats

Roosevelt Griffiths. 153 pages, 6x9 in. 88 illustrations. Red clothboard covers. Published by J. B. Lippincott Co., Philadelphia, Pa. Price $5.

Principles of operation embodied in various types of thermostats and temperature regulating instruments are described, and typical instruments are illustrated with diagrams and photographs. Although a few American designs are included, the text is a British publication and foreign instruments predominate.

Design principles for thermostats are given for types operated by gas pressure, expansion of liquids and mercury types. Applications of boiling liquids, expansion of solids and bimetal principles of actuation are also included, as well as electrical resistance and radiant energy types.

Throughout the text mechanical design principles, linkages and structures are given in detail. The chapter on bimetals lists various combinations and includes data for the design of bimetal elements.

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Dardelet Thread Handbook

220 pages, 4½x7 in. Many illustrations, charts and tables. Black cloth covers. Published by Dardelet Threadlock Corp., 120 Broadway, New York, N. Y. Price $2.

Although it contains general design data on threaded connections, the handbook deals principally with the theory and design of the Dardelet threadlocking construction. Test data, as well as thread dimensions and specifications, and thread forming operations are given. A final chapter of 55 pages contains useful tables on the proportions and weights of bolts, nuts and screws, including specifications and data for various materials.

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Illumination Course

206 pages, 8½x11 in. 170 illustrations. Paper covers. Published by the Westinghouse Lamp Co., Bloomfield, N. J. Price 50 cents.

While written as a textbook for a general course on illumination, information given in the first 80 pages on light sources, incandescent lamps and photometry will be of assistance to the designer. In this section, candle power, lumens and foot-candle are defined and described, and the visual properties of the eye are briefly discussed.

Other chapters in this section describe glare, reflection, transmission and refraction of light, production and visual properties of colored light. Standards of illumination for industrial interiors are given in a series of tables.

Selection of equipment for interior industrial and commercial lighting, display, advertising and exterior lighting make up the remainder of the book.

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Procedure Handbook of Arc Welding Design and Practice

450 pages, 5¾x9 in. Flexible binding. Published by the Lincoln Electric Co., Cleveland, Ohio. Price $1.50.

In this second edition of the book reviewed in November, 1933, new material has been included on weld inspection, photoelastic stress analysis, butt welds, design methods and welding of pipe lines.

Stethescopic, electromagnetic, X-ray and gamma ray inspection methods are briefly described in a new seven-page section. Another four-page addition describes photoelastic methods for determining stress concentration, and illustrates the effect of various types of welds on stress distribution. Detail drawings are included for an inexpensive mirror polariscope suitable for analyzing stresses.

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Design and Use of Instruments and Accurate Mechanisms

T. N. Whitehead. 283 pages, 5¾x8½ in. Blue clothboard covers. Published by The Macmillan Co., 60 Fifth Ave., New York, N. Y. Price $3.50.

Restricted to linear measuring instruments of the micrometer-microscope type, Part I of the text is an analytical treatment of the theory of errors, classifying errors into three groups: systematic, short period and erratic errors.

Design theories given in Part II indicate methods for controlling various types of errors in all precision instruments, although the examples used apply to astronomical or measuring microscopes.

Of particular interest is an analysis of design for stiffness showing the effect of damping and period of vibration on errors. Included in this chapter is a description of methods in use for isolation from temperature variations as well as vibrations.

Instruments described are all of foreign manufacture, text and style are typically British, and treatment is philosophical. But the fundamentals, of course, apply equally well for precision tools and measuring instruments of all types.

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Flame Temperatures in an Internal Combustion Engine

University of Illinois *Engineering Experiment Station, Bulletin No. 262,* by A. E. Hershey and R. F. Paton, 50 pages. Using a stroboscopic shutter and spectroscopic methods for measuring temperature, flame temperatures were found to vary from 3,750 deg. F. for rich or lean mixtures to a maximum temperature of 4,450 deg. F. for a 13.9 to 1 mixture. From instantaneous pressure measurements taken simultaneously with the flame temperatures, studies were made of the chemical reactions and equilibrium conditions at high temperatures. Diagrams and tables showing these relations indicate that the flame temperature is characteristic of thermal equilibrium during burning and subsequent expansion. Price 55 cents.

Effect of Heat-Treatment on the Expansivity of a Pyrex Glass

Bureau of Standards *Research Paper RP 626,* by J. B. Saunders and A. Q. Tool, 12 pages. Room temperature expansivity is shown to be varied as much as 10 percent by the annealing treatment. Expansion coefficients are reduced to 0.000,003,1 by annealing at 450 deg. C., higher temperatures causing higher coefficients.

Economic Lot Sizes in Manufacturing

Virginia Polytechnic Institute, Blacksburg, Va., *Engineering Extension Bulletin No. 31,* by P. T. Norton, Jr., 21 pages. This analysis brings out the importance of capital investment, deterioration of finished articles and storage charges in tabulating unit charges. Smaller lot sizes are shown to increase unit charges, while the capital expense increases for larger lots. Curves and charts show that the economic lot size varies approximately as the square root of the consumption rate.

A Study of the Locomotive Front End

University of Illinois, *Engineering Experiment Station Bulletin No. 256,* by Everett D. Young, 180 pages. Tests reported on a model of the locomotive "front end" or exhaust box show that plusating flow has little effect on the movement of air produced by the exhaust steam. The most important factors are the air resistance of the passages, the exhaust pressure, and details of the stack and nozzle. The report also includes a discussion of previous experiments and investigations.

Chemical and Physical Properties of Bronze Bearing Alloys

C. H. LEIS
Chief Engineer, Johnson Bronze Company

Johnson Bronze Alloy No.	Average Chemical Composition					Physical Properties								
	Copper Cu	Tin Sn	Lead Pb	Zinc Zn	Nickel Ni	Tensile lb. per sq.in. ± 5000	Proportional Limit ± 2000	Yield Point, lb. per sq. in. ± 3000	Elongation per cent in 2 in.	Brinell Hardness No.	(1) Wear Rate (dry)	(2) Coefficient of Friction (dry)	(3) Izod Notch Toughness	(4) Resistance to Pounding
19	70.0	11.0	19.0	...	...	27,500	7,800	20,700	8 ± 4	62	0.24	0.16	3.4	54
25	75.0	5.0	19.0	...	1.0	22,500	6,900	16,750	11 ± 4	44	0.36	0.14	5.2	22
27	80.0	10.0	10.0	...	...	30,000	9,700	19,000	10 ± 5	58	0.32	0.19	4.4	63
29	78.0	7.0	15.0	...	...	24,000	7,600	16,400	9 ± 4	52	0.35	0.16	5.6	40
51	87.0	10.0	1.0	2.0	...	36,500	12,500	18,500	15 ± 5	67	0.63	0.25	8.3	81
53	88.0	10.0	....	2.0	...	36,000	13,000	19,000	18 ± 4	69	0.62	0.26	8.5	86
55	86.0	12.0	....	2.0	...	39,200	13,600	21,000	10 ± 5	74	0.53	0.29	3.9	109
66	85.0	5.0	9.0	1.0	...	26,000	7,800	14,500	12 ± 5	48	0.50	0.19	8.4	20
71	85.0	5.0	5.0	5.0	...	29,000	8,200	16,500	20 ± 7	49	0.64	0.18	12.1	20
72	83.0	7.0	7.0	3.0	...	29,000	8,600	14,600	17 ± 5	56	0.41	0.19	8.6	38

(1) Wear rate in grams per 10,000 revolutions — Amsler wear test machine — without lubrication.

(2) Coefficient of friction as determined on Amsler wear test machine.

(3) Izod notch toughness, ft. lb. of work required to break specimen 0.400 × 0.400 in.

(4) Resistance to pounding, number of blows of hammer weighing 7.5 lb. falling 2 in. required to deform specimen 5 per cent. Specimen diameter 0.394 in., length 0.788 in.

Note: Other alloys may be found, whose chemical and physical characteristics differ but little, and consequently their performance does not materially enhance their bearing value. Any operating condition can be met by the above ten preferred bearing alloys. For a description of these alloys and their typical applications, see page 202 of this issue (June, 1934). On the following page is given a chart which indicates the field of application for bronzes of various percentages of copper, tin and lead.

Bearing Bronzes Grouped According to Fields of Use

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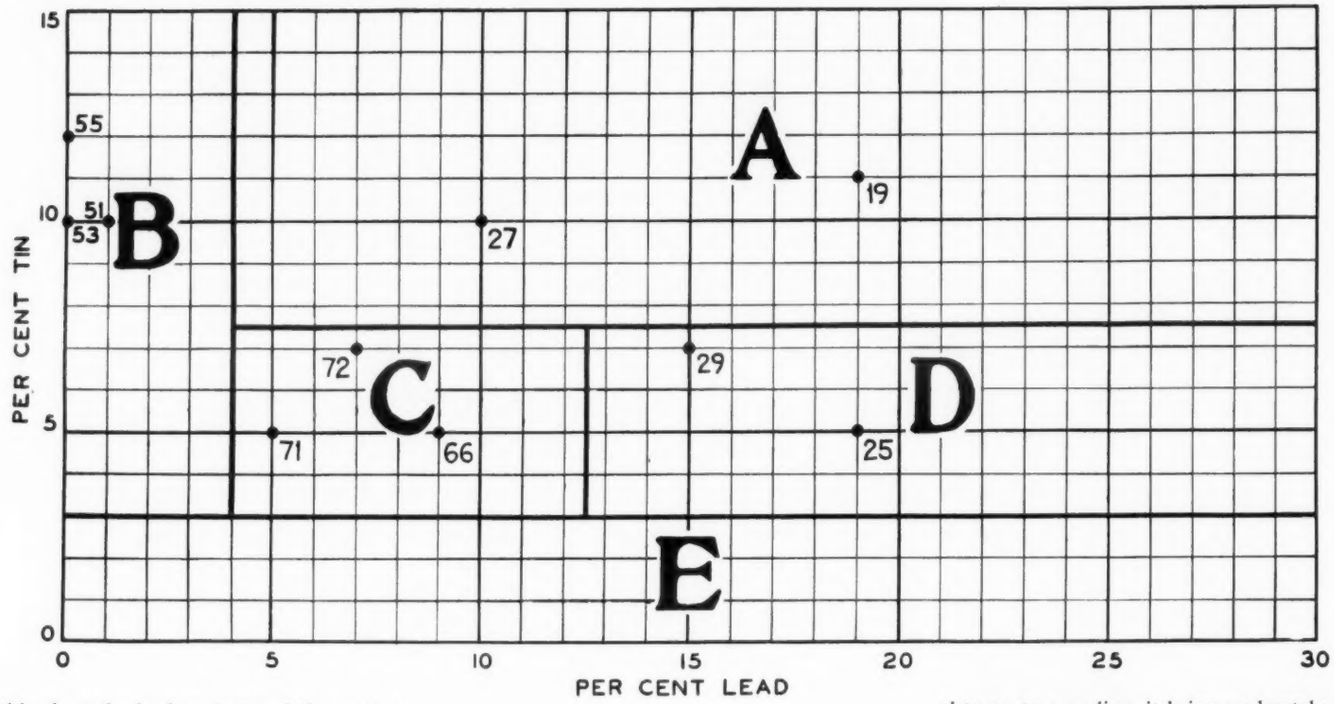

In this chart the lead content and tin content of the alloy is as designated by the coordinates. The per cent of copper content will be 100 minus the total of per cent of tin plus per cent of lead. The numbers in the field of the chart are the Johnson Bronze Alloy numbers.

On the preceding page will be found a table giving both the chemical compositions and physical properties of the alloys whose numbers appear in the field of this chart.

Refer to the article, "Bronze Bearing Alloys —Properties and Applications," beginning on page 202 of this number (June 1934) of *Product Engineering*, wherein is set forth the reasons for the various bearing requirements and the effect of each of the various constituents in copper—tin—lead alloys. The characteristics of the ten alloys included in the above chart and specific examples of their typical applications are also given.

Fields of Application of the Five Groups of Bronze Bearing Alloys

A—This is the most useful range of copper—tin—lead alloys. All alloys in this group have good wear rates and resistance to pounding. Alloy No. 19 has the highest wear rate and has comparatively good resistance to pounding, but is somewhat brittle. Alloy No. 27 has a good wear rate and a correspondingly good resistance to pounding, it being moderately tough.

B—Alloys in this group are suitable for bearing installations only where adequate lubrication can be guaranteed at all times. They have valuable characteristics where exceptionally heavy impact loads are encountered as in the bearings of crushing machinery.

C—In this group are the alloys best suited for low loads and moderately high speeds. These alloys are often used as bearing backs.

D—Alloys in this group are suiable for high speeds and low loads, but should not be used where there is excessive pounding.

E—Alloys in this group, containing less than 3 per cent tin, are unsuited for general bearing service owing to their high wear rate and low resistance to pounding.